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POWER

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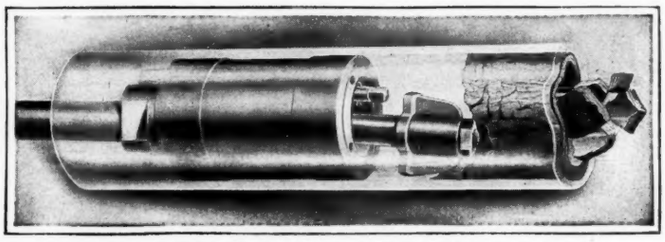

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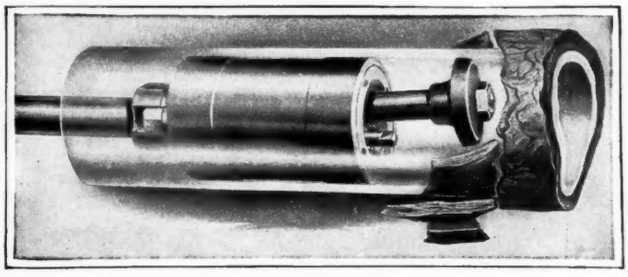

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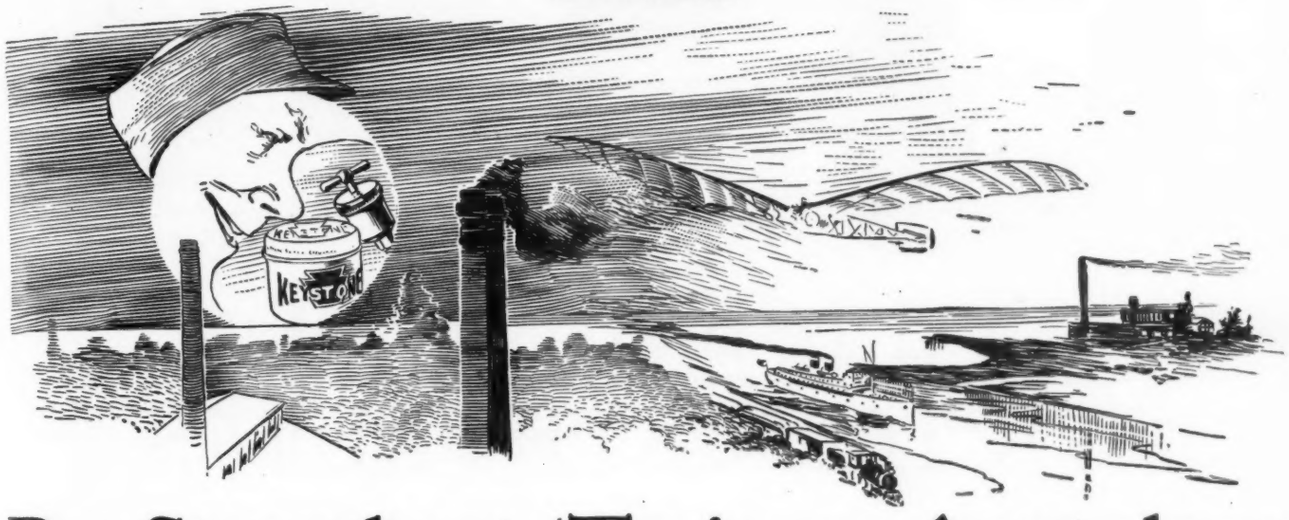

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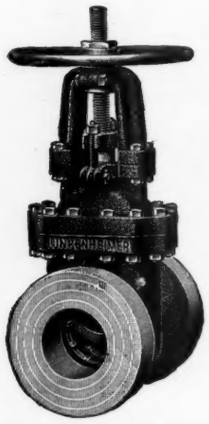

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POWER

Vol. 35 NEW YORK, JUNE 4, 1912 No. 23

A Fellow Engineer's Advice

IN a recent issue of POWER we hear the old cry of engineers being underpaid. Especially among the old timers there also appears to be a prejudice against college graduates who are rapidly filling positions formerly held by practical men. Why this condition exists is evident to anyone who has noticed the development of power plants during the past few years. The plant owner of today will not pay a good salary for the old rule of thumb methods which answered some years ago. He can get a college man with scientific training who may lack practical knowledge at first, but is likely to lower the plant's cost more than enough to offset this.

On this page in the last issue we urged the college graduate to enter the power plant as the best place to get the training and practical experience he needs. Until he does this the practical engineer has the advantage of him. That many are now doing this seems to be taken by some operating engineers as cause for alarm—a menace to their welfare.

Such is not the case, for, as we have said, the operating engineer can easily hold his own if he is equally ambitious and improves his opportunities to add to his theoretical knowledge by study and experimentation.

A letter recently received from an operating engineer takes such a sensible view of the situation regarding the college man's entering his field, and the duties of engineers to their employers, that we print it here in full.

Now instead of howling about the pay and carrying a chip on our shoulder for the collegian to knock off, it appears to me that we had better get busy and equip ourselves with the knowledge that will enable us to command better pay and hold our own with the college graduate. Most employers, if properly approached, will pay a man his price if it is to their interest to do so. It is up to us to show him just where and how it is and we must prove conclusively with facts and figures, not mere words, that they are not wasting but saving money.

Many owners of today are getting acquainted with the indicator and some can take and read the diagrams although they are not engineers. This type of owner naturally expects his engineer to do the same. In fact every engineer should be master of the indicator, but unfortunately many are not. The same is true of combustion and flue gas analysis. Modern boiler plants are no longer run on guess work. Scientific management replaces rule of thumb methods. Efficiency is the watch word.

Lack of financial means is no longer a bar to the man who makes up his mind to acquire knowledge. There are numerous ways of acquiring the knowledge which will, if he persists, enable him to command a better salary and which will eventually raise the standard of the engineer at large. There is always work for the man who is above the average. It's the average man that suffers.

Calculating the Flow of Water in Pipes

By T. W. Holloway

General formulas with illustrative examples for the velocity of flow, quantity of discharge, head required to produce a given delivery, diameter of pipes to meet specified conditions and the comparative flow of water in pipes of different sizes.

In power-plant work, hydraulic problems are continually coming up and the engineer is frequently called upon to make calculations pertaining to the flow of water in pipes. This is a subject which can be made very extensive, especially where accurate results are required, but for practical calculations, such as are usually required around the power house, it is not necessary to go into a deep theoretical study of the subject.

The quantity of water passing any section of a pipe of uniform diameter in a given time will depend upon the cross-sectional area of the pipe, and the velocity at which the water is moving.

Let

a = Area of cross-section of the pipe in square feet;

v = Velocity of flow in feet per second;

q = Number of cubic feet of water passing a given section in one second;

then,

$$q = a v$$

For example: If water is flowing through a 6-inch pipe at the rate of 10 feet per second and the quantity delivered per second is required, it is necessary only to multiply the area of the cross-section of the pipe by the velocity. The area of a pipe is

$$\tfrac{1}{4}\ \pi\ d^2 = 0.7854\ d^2$$

where d = the diameter of the pipe in inches. Therefore, the quantity in cubic feet passing a given section is

$$q = 0.7854 \times (\tfrac{6}{12})^2 \times 10 = 1.963$$

or approximately 2 cubic feet per second. If the quantity is desired in gallons, then multiply the discharge in cubic feet by 7.48. If it is desired to reduce gallons to pounds, as is sometimes necessary, multiply the discharge in gallons by 8⅓. To reduce cubic feet to pounds, multiply by 62½.

The theoretical velocity—that is, the velocity the water would have if there were no resistance due to friction in the pipe, resistance of bends, valves, etc.—would be the same as the velocity the water would attain in falling through a hight equal to the head. Expressed as a formula,

in which

v = Velocity, in feet per second;

g = Acceleration due to gravity = 32.16;

h = Head, in feet.

By the head, which causes the flow, is meant the difference in level from the surface of the water in the reservoir to the center of the outlet, when the discharge occurs freely into the air, as shown in Fig. 1.

The theoretical velocity may be quite different from the actual velocity if the pipe is of great length, or if there are bends or valves to increase the frictional resistance in the pipe; but the theoretical amount often serves as a guide in making calculations. Suppose that it is necessary to find the velocity of flow and the discharge from a 6-inch pipe when the head is 50 feet.

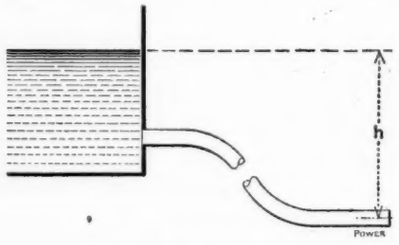


FIG. 1

$$v = \sqrt{2gh} = \sqrt{2 \times 32.16 \times 50} = 57 \textit{ feet per second}$$

The quantity in cubic feet per second will be

$$q = a v = 0.7854 \times (\tfrac{1}{2})^2 \times 57 = 11.19 \textit{ cubic feet}$$

Instead of the head being given in feet it is frequently expressed in pounds per square inch, as in the case where a head is furnished by a pump. This being the case, the head in pounds may be changed readily to head in feet by dividing by 0.434, or, what is the same thing, multiplying by 2.31. This is because a column of water 1 foot in hight exerts a pressure of 0.434 pound, or a column of water 2.31 feet in hight exerts a pressure of 1 pound.

If w represents the number of pounds of water discharged per second, the theoretical work which the falling water is capable of doing will be $w\,h$ foot-pounds, because work is the product of the force and the distance through which the force acts. In this case the force is the weight of water w, and the distance is the head h.

The kinetic energy of the flowing water is $\frac{w v^2}{2g}$, and if no work is expended in overcoming resistance due to friction or other losses, then the expression for work and for kinetic energy must be equal, or

$$w h = \frac{w v^2}{2g}$$

From which

$$h = \frac{v^2}{2g}$$

In actual practice, especially in the case of long pipes, the kinetic energy is less than the theoretical work, the difference being due to the energy lost in overcoming friction. Consequently, $\frac{v^2}{2g}$ in practice is less than the total head h. Now, $\frac{v^2}{2g}$ is the effective head, corresponding to the velocity at discharge, and is sometimes spoken of as the velocity head. Therefore the loss of head due to frictional resistance must equal

$$h - \frac{v^2}{2g}$$

For great lengths of pipe the velocity may become very small, showing that nearly all of the head has been lost in overcoming the resistance.

Several conditions must be taken into account to find the actual velocity of water in a pipe, the number of conditions depending on the particular pipe line under consideration. When water from a

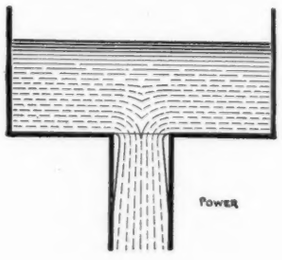


FIG. 2

reservoir, tank or other source of supply enters a pipe there is more or less contraction and resistance of the stream, as shown in Fig. 2, the amount depending upon the manner of connection.

The loss of head due to contraction and resistance of the inner edges of the pipe is known as the loss of entrance, and depends on the manner of connection and the shape of the end of the pipe. The loss of head may be expressed by the formula

$$h_1 = m \frac{v^2}{2g}$$

in which

h_1 = Head, in feet, lost at entrance;

v = Velocity, in feet per second;

m = Coefficient depending on the form of the end of the pipe.

When the pipe projects into the reservoir or tank m equals 0.93; when it is flush with the reservoir, as in Fig. 2, m may be taken as 0.5, and when the inner end of the pipe is provided with a bell-shaped mouth m equals 0.

Loss of head due to friction and cohesion along the interior surface of the pipe must be taken into account, as this is usually large, especially in long pipes. A formula for the head lost in frictional resistance which has been derived from experimental data is as follows:

$$h_2 = f\frac{v^2 l}{2gd}$$

in which

h_2 = Head lost by friction;

f = Friction factor which depends on the condition of the interior of the pipe;

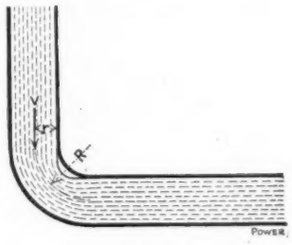


FIG. 3

l = Length of pipe in feet;

v = Velocity of water, in feet per second, flowing in pipe;

g = 32.16;

d = Diameter of pipe in feet.

The value of f ranges from 0.05 to 0.01 for new clean pipes, but for the average conditions in practice f is taken as 0.02, which value may be used in making calculations unless another value is specified.

If a pipe is straight and of uniform size throughout, the loss of head at the entrance and that due to frictional resistance of the interior surface is all that need be considered, but if the pipe varies in diameter, has sharp bends or turns, or contains valves, other losses must be considered.

In practice, variation in diameter of the pipe is usually made by means of a reducer so that the change in section is gradual. This being the case, the loss of head due to this cause is very small, and for most practical purposes may be neglected.

Sharp bends or turns, as in Fig. 3, cause considerable loss of head, especially when several such abrupt changes are made. The loss of head caused by a sharp bend of 90 degrees, as in an elbow, may be found by the following formula:

$$h_3 = c\frac{v^2}{2g}$$

in which

h_3 = Loss of head in feet;

c = Constant which depends on the ratio $\frac{r}{R}$, of the inside radius of the pipe to the radius of curvature of the bend;

v and g are the same as in previous formulas.

The values for c may be found from Table 1, in which r is the radius of the pipe and R the radius of the bend.

Valves for regulating the flow in pipes may cause considerable loss of head when they are not fully opened. The loss of head may be found by the formula,

$$h_4 = c'\frac{v^2}{2g}$$

The constant c' for a cock valve, as shown at the left in Fig. 4, may be found from Table 2, in which a is the angle which the opening through the valve makes with the axis of the pipe.

TABLE 1. COEFFICIENT FOR LOSS OF HEAD DUE TO SHARP BENDS

$\frac{r}{R}$ =	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
c =	0.13	0.14	0.16	0.21	0.29	0.44	0.66	0.98	1.41	1.98

TABLE 2. COEFFICIENT FOR LOSS OF HEAD DUE TO COCK VALVES

a =	0°	10°	20°	30°	40°	50°	55°	60°	65°
c' =	0	0.29	1.6	5.5	17	53	106	206	486

The constant c' for a gate valve, as indicated at the right in Fig. 4, may be found from Table 3, in which $\frac{d_1}{d}$ is the

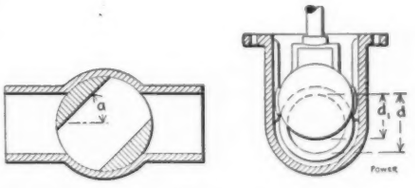


FIG. 4

ratio of the distance the gate is lowered below the top of the pipe, to the diameter of the pipe.

VELOCITY OF FLOW

Having determined the losses of head, due to various conditions, the resulting velocity may now be found. Since h loss of head, $h = \frac{v^2}{2g}$ must equal the velocity head, or the head corresponding to the velocity of the issuing stream, the lost head, $h = \frac{v^2}{2g}$, must equal the sum of the various head losses; that is,

$$h - \frac{v^2}{2g} = h_1 + h_2 + h_3 + h_4$$

or

$$h - \frac{v^2}{2g} = m\frac{v^2}{2g} + f\frac{v^2 l}{2gd} + c\frac{v^2}{2g} + c'\frac{v^2}{2g}$$

Dividing this expression by $\frac{v^2}{2g}$,

$$\frac{2gh}{v^2} - 1 = m + f\frac{l}{d} + c + c'$$

from which

$$v = \sqrt{\frac{2gh}{1 + m + f\frac{l}{d} + c + c'}} \qquad (1)$$

which is a general formula that may be used for the velocity of flow. If there are no valves to be taken into account, then c' becomes zero, and if there is more than one valve, then c' represents the sum of the constants for the various valves. In a similar manner, if there are no bends in the pipe c becomes zero, and if there is more than one bend, c represents the sum of the constants for the various bends.

Example: A 6-inch pipe line 500 feet in length delivers water from a reservoir 60 feet above the outlet. The pipe makes two right-angled turns and the gate valve at the reservoir is wide open. If the pipe projects into the reservoir and the radius of the bends is 15 inches, what is the velocity of flow in the pipe?

Solution: Apply the general formula (1) for the velocity of flow, substituting the following values: g = 32.16; h = 60 feet; m = 0.93; f may be taken as 0.02; l = 500 feet; d = ½ foot. Since the radius of the bends is 15 inches, then from Table 1

TABLE 3. COEFFICIENT FOR LOSS OF HEAD DUE TO GATE VALVES

$\frac{d_1}{d}$ =	0	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$
c' =	0	0.07	0.26	0.81	2.1	5.5	17	98

$$\frac{r}{R} = \frac{3}{15} = 0.2$$

and c = 0.14 for one bend or 0.28 for two bends; c' = 0, since the gate valve

is wide open. Substituting in the formula,

$$v = \sqrt{\frac{2 \times 32.16 \times 60}{1 + 0.93 + 0.02 \times \frac{500}{0.5} + 0.28 + o}}$$

$$= 13 \textit{ feet per second}$$

Quantity of Discharge

The general formula for the velocity of flow in feet per second having been obtained, the discharge, q in cubic feet per second, is readily found from the expression

$$q = a\,v$$

previously given. This becomes

$$q = 0.7854\,d^2 \sqrt{\frac{2\,gh}{1 + m + f\frac{l}{d} + c + c'}} =$$

$$6.3\,d^2 \sqrt{\frac{h}{1 + m + f\frac{l}{d} + c + c'}} \qquad (2)$$

In order to show the application of the foregoing formulas, a practical problem which recently came up will be solved.

A tank 20 feet in hight and 20 feet in diameter is erected on a tower 30 feet high. Into the center of the bottom of the tank a 6-inch pipe is connected so that the pipe is flush with the bottom of the tank. The pipe runs 30 feet vertically downward, then turns at 90 degrees, running horizontally 300 feet and discharging through a gate valve, as shown in Fig. 5. What is the discharge when the tank is full?

The total head in this case is

$$30 + 20 = 50 \textit{ feet}$$

and the total length of the pipe will be about

$$300 + 30 = 330 \textit{ feet}$$

In this case there will be loss of head at the entrance to the pipe, loss of head due to friction in the pipe, loss of head due to the 90-degree bend and loss of head due to the gate valve if it is not fully opened. In this case, however, the gate valve has been assumed as wide open. Applying the general formula and substituting for the values of the various factors: $g = 32.16$; $h = 50$ feet; $m = 0.5$, since the end of the pipe is flush with the bottom of the tank; $f = 0.02$; $l = 330$ feet; $d = \frac{6}{12} = \frac{1}{2}$ foot; $c = 0.29$, as found from Table 1 if the 90-degree bend has a radius of 6 inches; as the gate valve is assumed to be wide open, c' becomes zero.

$$= 6.3 \times (\tfrac{6}{12})^2 \sqrt{\frac{50}{1 + 0.5 + 0.02 \times \frac{330}{0.5} + 0.29}}$$

$$= 2.87 \textit{ cubic feet per second}$$

Expressed in gallons, the discharge is

$$2.87 \times 7.48 = 21.47$$

or approximately 21½ gallons. This discharge is on the assumption that the gate valve is wide open. If the gate valve were partly closed it would be necessary to know the ratio $\frac{d_1}{d}$, Fig. 4, and then substitute the value for c', as found from Table 3.

Head Required to Produce a Given Delivery

It is sometimes necessary to find the head required to produce a given flow. This being the case, the general formula (2) for the quantity of discharge may be transposed and the value of h found. Squaring both sides of the equation and solving for h, a general formula for the required head is

$$h = 0.0252\,(1 + m + f\frac{l}{d} + c + c')\frac{q^2}{d^4} \qquad (3)$$

TABLE 4. SIZES OF STANDARD PIPES

Size, Inches	Diameter		Size, Inches	Diameter		Size, Inches	Diameter	
	Internal	External		Internal	External		Internal	External
⅛	0.27	0.40	2½	2.47	2.87	9	8.94	9.62
¼	0.36	0.54	3	3.07	3.5	10	10.02	10.75
⅜	0.49	0.67	3½	3.55	4	11	11	11.75
½	0.62	0.84	4	4.03	4.5	12	12	12.75
¾	0.82	1.05	4½	4.51	5	13	13.25	14
1	1.05	1.31	5	5.04	5.56	14	14.25	15
1¼	1.38	1.66	6	6.06	6.62	15	15.43	16
1½	1.61	1.90	7	7.02	7.62	16	16.4	17
2	2.07	2.37	8	7.98	8.62	17	17.32	18

Diameters of Pipes to Meet Specified Conditions

In practice it is often necessary to find the diameter of the pipe to give a required discharge. A general formula suitable for this purpose may be derived from formula (3), but this would run into some difficult mathematics, and in order to find the diameter it would be necessary to extract the fifth root, which would require a knowledge of logarithms. A simpler method, and one which is just as satisfactory, is to assume a diameter and then substitute this assumed value in either formula (2) for finding the quantity of discharge, or in formula (3) for finding the head. If the first assumption made does not give the quantity of discharge or the head desired, assume a new value for the diameter and repeat the calculation. By this cut-and-try method a diameter may be found which will satisfy the conditions. This diameter will then be taken as the diameter of the pipe.

For example: Let it be required to find the diameter of a pipe to give a discharge of 21½ gallons per second for the conditions in Fig. 5. The discharge in cubic feet is

$$21.5 \div 7.48 = 2.87$$

As a trial value, assume the diameter of the pipe to be 5 inches or 0.4166 foot. Taking a value of $m = 0.5$, $f = 0.02$, $l = 330$ feet, and if the radius of the bend is taken equal to the diameter of the pipe or the ratio of the radius of the pipe to the radius of the bend equals 0.5, then from Table 1 the constant $c = 0.29$ and $c' = 0$. Substituting in formula (3) and solving for the head,

$$h = 0.0252\,(1 + 0.5 + 0.02 \times \frac{330}{0.4166} +$$

$$0.29 + 0)\frac{(2.87)^2}{(0.4166)^4} = 121.48 \textit{ feet}$$

which is too great. Since the head is to be 50 feet, a larger size of pipe must be used. Assuming a 6-inch pipe and recalculating,

$$h = 0.0252\,(1 + 0.5 + 0.02 \times \frac{330}{0.5} +$$

$$0.29 + 0)\frac{(2.87)^2}{(0.5)^4} = 50 \textit{ feet}$$

which is the head required. Therefore, a 6-inch pipe would be used.

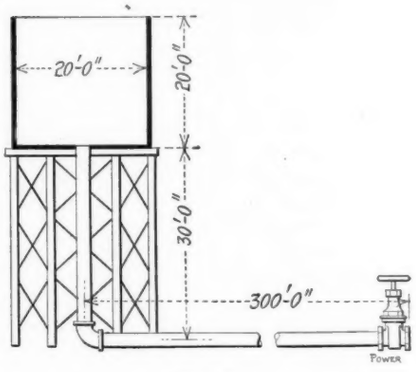


FIG. 5

Comparative Flow of Water in Pipes of Different Sizes

The foregoing example shows that in the particular case in question the 5-inch pipe requires more than twice as much head as the 6-inch pipe in order to give the same quantity of discharge. Of course, the velocity must be much higher in the 5-inch pipe. At the same velocity of flow two pipes discharge as the square of their diameters—that is, in the case of the pipes mentioned the ratio of discharge will be $(5)^2 : (6)^2$, or 25 to 36. The same head will not produce the same velocity in pipes of different diameters or lengths. The chief resistance to the flow of the water in a pipe is due to the friction of the water upon the sides of the pipe. This friction is greater in proportion to the contents of a small pipe than of a large pipe—that is, in a given length of pipe there is more interior surface in proportion to the contents in the

case of the small pipe than in a larger pipe, and consequently the frictional resistance is greater in the small pipe.

The relative flow in pipes of different sizes under similar conditions may be found approximately by the following proportion, which has been derived from a large number of experiments:

$$N : N_1 = \frac{d_1^3}{\sqrt{d_1 + 3.6}} : \frac{d^3}{\sqrt{d + 3.6}}$$

in which N is the number of pipes of diameter d and N_1 is the number of pipes of diameter d_1.

For example, suppose it is necessary to find the number of 2-inch pipes required to replace one 7-inch pipe so that the quantity of discharge under similar conditions in each case will be the same. The actual internal diameter of a standard 2-inch pipe is 2.07 inches and of a 7-inch pipe is 7.02 inches. Substituting in the proportion:

$$1 : N_1 :: \frac{(2.07)^3}{\sqrt{2.07 + 3.6}} : \frac{(7.02)^3}{\sqrt{7.02 + 3.6}}$$

or

$$1 : N_1 = 3.64 : 106.18$$

from which

$$N_1 = \frac{106.18}{3.64} = 29 \text{ pipes}$$

That is, it requires twenty-nine standard 2-inch pipes to equal one standard 7-inch pipe.

Standard pipe diameters vary slightly from the nominal diameters, as is shown by Table 4.

This table will be found convenient, especially where accurate work is required, and it is necessary to use actual diameters instead of nominal diameters.

The foregoing standard formulas have been found to give good results in practice, but exact calculations are impossible in a majority of hydraulic examples. Actual results may vary from 0 to 10 per cent. from the calculated results, but this is as close as can be expected, due to the number of different factors which tend to modify the flow of a liquid.

The De Laval Multi-Stage Turbine

The turbine invented by Dr. De Laval and made by the several companies bearing his name is of the simplest type, in which the steam is utilized in a single expansion from the initial to the exhaust or condenser pressure. The jets impinge with high velocity upon a single row of buckets, the rotative speed of which is limited by centrifugal force. Thus reduction gearing is required to adapt the turbine to most uses. The maximum wheel diameter of single-stage turbines is about 30 in. and the horsepower about 700.

The turbine is supplied with or without the double helical reduction gears which have been a familiar feature of De Laval turbines, and is adapted to various steam conditions.

It is steam sealed, segmental carbon rings being used in addition to the usual labyrinth packing.

It was to be expected that a pioneer company in this field would progress with the advance of turbine engineering and develop a unit of more complex design and adapted to greater capacities and higher efficiencies.

Some time ago POWER described some large multistage units built by the De Laval Co., of Sweden, and the American company, of Trenton, N. J., now announces its ability to furnish units of this design. The type adopted is the multipressure stage turbine.

Although this unit is supplied without gearing, normally it is furnished to drive through twin helical gears. This allows the best design of steam end irrespective of the speed of the driven machine, and

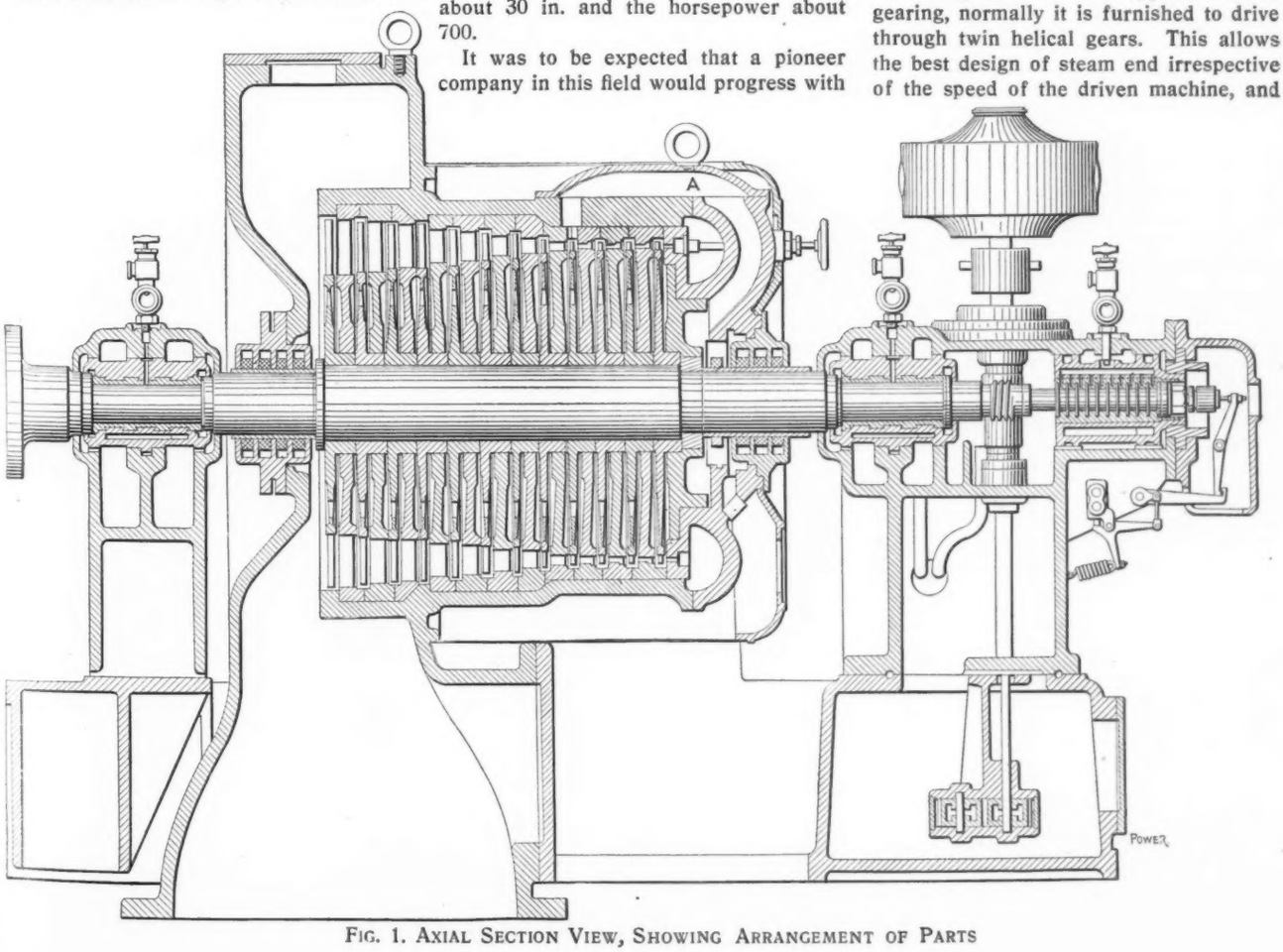


FIG. 1. AXIAL SECTION VIEW, SHOWING ARRANGEMENT OF PARTS

of easily connecting the unit to existing shafting; it also adapts it to direct-current generators or belt and rope drive.

Partial admission to the first chamber is through hand-controlled nozzles which may be adjusted to suit load conditions; full admission to all chambers occurs thereafter, the increase in the area of the steam passages being obtained by lengthening the blades. Fig. 1 is an axial section showing the general arrangement of the turbine. The shaft is heavy and the rated number of revolutions well within the critical speed. The hubs of the wheels extend flush with those of adjacent ones, which adds to the stiffness of the shaft. This design permits them to be locked by one nut, and as taper bushings are used in mounting them on the shaft they may be easily removed.

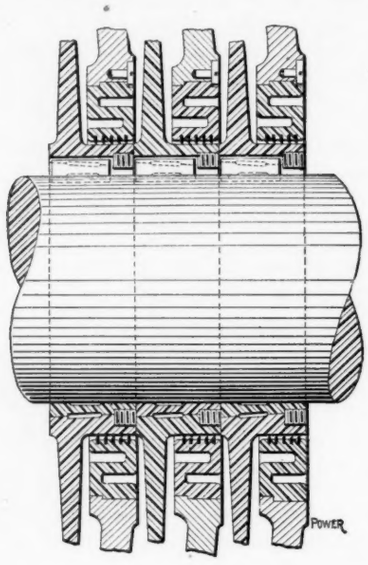


FIG. 2. SHOWING LABYRINTH PACKING

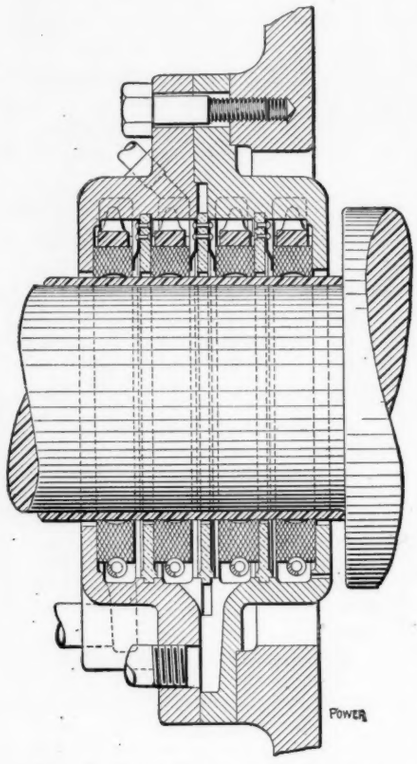


FIG. 3. SHOWING CARBON PACKING

Excepting the nozzles in the first stage, those of each succeeding stage are formed between guide vanes set around the entire periphery of the diaphragms. The diaphragm disks are perforated at the center and fitted with removable labyrinth packing which reduces the leakage from stage to stage between the wheel hubs and the diaphragm. Leakage from the first cell outward to the atmosphere is prevented by labyrinth packing, as shown in Fig. 2, and steam leaking past this packing is led back to an intermediate stage through the passage *A*, Fig. 1.

Following the labyrinth packing is a series of segmental carbon ring packings, each ring being inclosed in a separate compartment, as shown in Fig. 3. The leakage past the first carbon ring is carried back to the exhaust connection or to an intermediate stage of the low-pressure packing. Referring to Fig. 3, provision is made for introducing live steam at reduced pressure between the second and third rings, while on the steam end it may be admitted between the two outermost rings to seal against air leakage into the turbine.

There are three bearings, two being main bearings supporting the revolving parts and the third an adjustable marine-type thrust bearing. The axial thrust, however, is slight as each wheel is surrounded by steam of practically uniform pressure. Each bearing is arranged for water cooling. Lubricating oil is supplied by a circulating pump of the positive-gear type, driven from the lower end of the governor spindle. This pump draws the oil from a reservoir in the box-type bedplate through filter screens, and forces it to an elevated tank from which it gravitates to the sight-feed lubricator on each bearing.

The wheel case is split horizontally in the usual manner, and provision is made for radial expansion by supporting the case in the plane of the center line upon two pedestals rising from the bedplate.

The speed of the turbine is throttle-regulated by a governor of the Jahns type, the valve of which is of the double-seated, balanced-poppet type. The usual safety governor operating an independent valve is also provided.

Fig. 4 shows the turbine driving a 1500-kw. alternator at 1500 r.p.m.

Heating and ventilating engineers welcome the steam turbine for back-pressure service—that is, where the exhaust steam is to be utilized for heating or drying or other commercial purposes—because there is no oil in the exhaust. The absence of pulsations, vibrations and other undesirable qualities also renders this type ideal for all services where power is a byproduct of a steam-using system, such as heating or drying systems.

The trouble with most commercial turbines has been their high steam consumption; consequently where heating is required only during certain months, the great waste of steam during the warm season has about over-balanced the advantage gained by the double utilization of the steam during the heating season. Aside from the unavoidable inefficiency of the high-pressure stages of the reaction type of turbine, due to leakage, this unfavorable opinion has been based in the case of turbines of other types, largely upon experience with turbines built up of large wheels originally intended for condensing service. The loss of power from the friction of a disk running at constant peripheral speed increases as the square of the diameter of the disk and also as the density of the steam; therefore in dense steam as small wheels should be used as is consistent with the flow area necessary to develop the power desired.

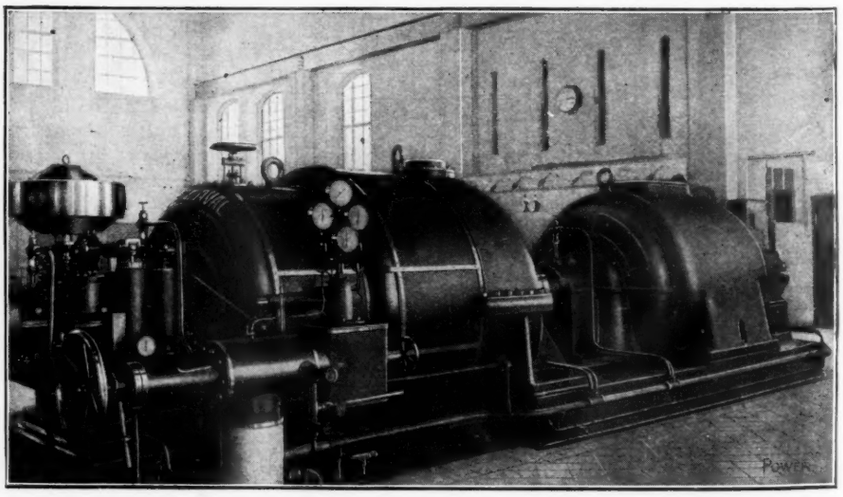


FIG. 4. TURBINE DRIVING 1500-KW. ALTERNATOR

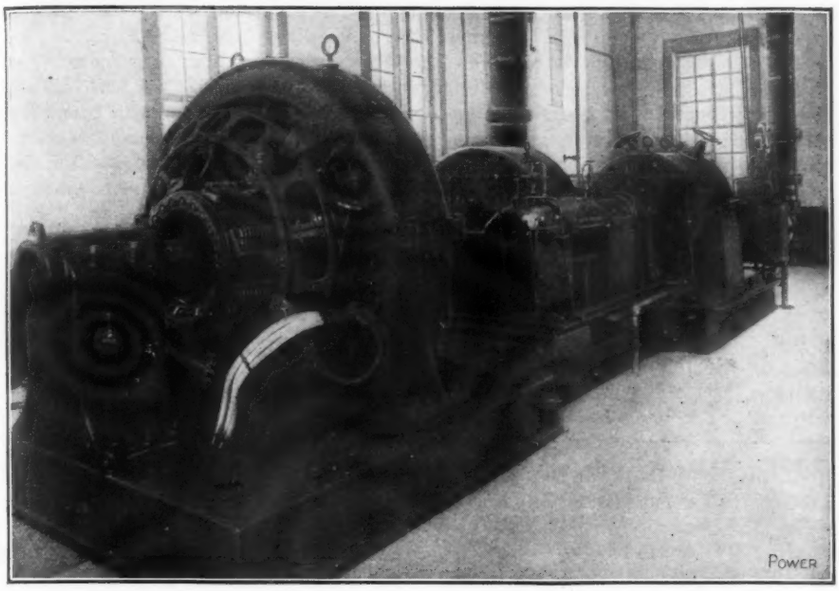


FIG. 5. GEARED TURBINE DRIVING DIRECT-CURRENT GENERATOR

However, with a given steam velocity and the proper ratio of bucket to steam velocity—that is, with buckets moving about one-half as fast as the steam in impulse turbines—small disks lead to high speeds of rotation, which are unsuitable for driving machinery direct-connected. To overcome this difficulty the De Laval turbine is equipped with the helical gearing referred to and described in POWER in the issue of Apr. 11, 1911.

Fig. 5 shows a turbine of this type installed in a cloth-finishing establishment. The plant is old and the power equipment has grown by various additions from time to time. Originally, waterwheels were installed, driving shafting, and later on reciprocating engines were belted to the same shafting. Part of the transmission through the mill is by belt and part by current from generators belt-driven from the shafting. Direct current is used to secure the advantages of variable-speed motors for the printing machines.

The steam turbine was originally put in to help out during the dry season, and the steam required for washing, finishing, bleaching and drying operations was taken at such times through reducing valves. Eventually, it was seen that it would be more economical to allow the turbine to run continuously, supplying the steam required for manufacturing operations, and to shut down the engine, which was operated condensing and the exhaust of which was therefore lost. This conviction was strengthened by the fact that the turbine showed a high thermodynamic efficiency, realizing some 70 per cent. of the total work available from steam between boiler pressure and the back pressure in the exhaust line.

This exceptionally good performance is due both to the small wheels and the high speed used, and also to the close clearances between the shaft and diaphragms made possible by a comparatively small number of wheels and a stiff, rigid shaft. The turbine runs at 3600 r.p.m. and drives a 400-kw. Crocker-Wheeler generator at 600 r.p.m., generating current at 250 volts.

San Antonio Locomotive Explosion*

In its issue of Apr. 9, POWER published an account of the San Antonio locomotive-boiler explosion occurring at 9:30 on the morning of Mar. 18. The report on the accident by Chief Inspector Ensign to the Interstate Commerce Commission, recently received, is so interesting that it is here printed, practically in full.

An incorrect gage and tightening the adjusting screws of the safety valves permitted a pressure beyond endurance of the boiler.

Using 1-in. instead of 1¼-in. bolts for the sling stays reduced the factor of safety to 2.67.

*Abstract of report by John F. Ensign, chief inspector of locomotive boilers, to Interstate Commerce Commission.

BOILER CONSTRUCTION

The locomotive was of the heavy passenger 4-6-0 type, using crude oil for fuel, and was owned and operated by the Galveston, Harrisburg & San Antonio Ry. Co.; it was built in March, 1908, by the American Locomotive Co. at the Brooks Works. The firebox was of three-piece construction, crown-bar type. The working steam pressure was 200 lb. per square inch. The barrel of the boiler was made of ¾-in. steel in three sections or courses, constructed with butt longitudinal joints having diamond-shaped welts. The dome was located on the third course. The wrapper sheet was 5/8-in. steel, the back head sheet and the back flue sheet ½-in. steel, and the firebox-door sheet, crown- and side sheets ⅜-in. steel. The firebox was stayed with rigid bolts ⅞ in. diameter at the ends, reduced to ¾ in. at the center of the bolts; four rows of 1-in. Tate flexible bolts at the top of the firebox and two rows at each end, staggered at the top corners. The crown bolts were of a driving fit with countersunk heads, 1⅛ in. diameter at the bottom end and 1 in. diameter at the top end, extending through the crown bars with nuts on the top. The crown-sheet was supported by 15 crown bars hung from the wrapper sheet by 168 sling stays, ⅝x3 in., and 12 sling stays ½x2¾ in. The flues, numbering 355, were of 2 in. diameter. The boiler was equipped with three 3-in. Crosby safety valves.

SUMMARY OF EVIDENCE

Briefly summarized, the investigation conducted by the railroad commission of Texas brought out the following facts: The locomotive was out of service from Feb. 21 to Mar. 18, 1912, for repairs, when the following boiler work was done: Two hundred flues reset, one back head brace repaired, one front flue-sheet brace and two throat-stays repaired, eight stay-bolts renewed, safety valves ground in, steam gage tested and hydrostatic test of 250 lb. pressure per square inch applied. Repairs were completed about 5:45 p.m., Mar. 17, and the locomotive fired up, but no steam was raised. It was again fired up about 6:10 a.m., Mar. 18. The safety valves first opened at 7:30 a.m., at which time the steam gage registered 50 lb. pressure. The safety valves were screwed down and again opened at about 8 a.m., when the gage registered 150 lb. pressure. There is no evidence that the safety valves opened at any time subsequent to 8 a.m. The locomotive had a heavy forced-oil fire from 8 to 8:55 a.m., at which time the explosion occurred.

A railroad employee was engaged in setting the safety valves at the time of the explosion, and, although the evidence showed that the gage had been tested, there was no evidence that the siphon pipe leading from the gage to the

boiler had been cleaned between the valve and the boiler, which is the point where it would be most likely to be obstructed; nor does it appear that the valve was open.

The damage to the boiler, as well as the direction in which the various parts

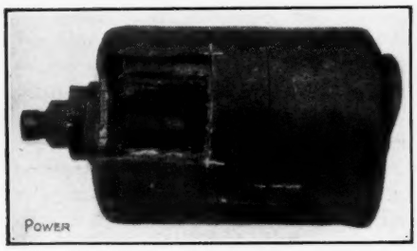


FIG. 1. SAFETY-VALVE CASING, SHOWING ABSENCE OF LOCKNUT AND BENT CONDITION OF SCREW

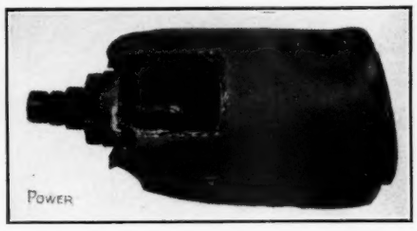


FIG. 2. SHOWING DISTANCE ADJUSTING SCREW WAS TURNED DOWN TO LOCKING NUT

FIG. 3. LOCKING NUT ABSENT AND ADJUSTING SCREW TURNED DOWN AS FAR AS POSSIBLE

of the boiler were thrown, indicate conclusively that the firebox sheets were the first to give way, as the boiler head was blown backward and all other parts of the boiler forward, the flues, flue sheets and smoke arch being simply turned over forward and thrown to the left, while the wrapper sheet and a part of the dome course with the dome attached and other pieces of the shell sheets were driven for long distances forward and to the right.

CONDITION OF SAFETY VALVES

Owing to the damaged condition of the safety valves and the inability to recover the springs and valves, a test thereof could not be made. The casings with the adjusting screws were found, one of which had no locknut. The hexagon-shaped heads on the adjusting screws had the corners twisted off, after which a Stillson wrench had apparently been used in an effort to screw them down further.

One of the adjusting screws was bent and the bottom end was upset or burred by the pressure put on it; the condition of the threads on the adjusting screws indicated that they had been recently screwed down more than 5/8 in. Reasonable knowledge of the purpose and construction of safety valves should prevent work being done on them which would cause such conditions. Views of the safety-valve casings sectioned to show the condition of the adjusting screws are shown in Figs. 1, 2 and 3.

FAILURE OF CROWN-BAR SLING STAYS

A careful examination of the crown-bar sling stays showed that they were made of wrought iron, while the specifications called for steel. The sling stays were badly stretched and reduced in section at the eyes where they failed, as shown in Figs. 4 and 5, indicating a gradual rise of pressure in the boiler.

Five 1-in. bolts were used to attach the sling stays to the crown bars and to the wrapper sheet, where 1¼-in. bolts should have been used; they were specified on the drawings, except on the front crown bar, which called for 1-in. bolts. The crown bars in this boiler were not supported on the sides of the firebox, as was customary in the older type of crown-bar boilers; therefore all the support was from the sling stays.

Five crown-bar sling stays from the locomotive were tested by the United States Bureau of Standards to determine the load the stays would support when 1-in. and 1¼-in. bolts were used. The bolts used in making these tests were those used in the boiler at the time of the explosion. Two stays, Nos. 1 and 2, tested with 1-in. bolts, failed with a total load of 26,650 and 21,840 lb. respectively. Three stays, Nos. 3, 4 and 5, tested with 1¼-in. bolts, failed with a total load of 30,000, 33,890 and 31,620 lb. respectively.

Using 21,840 lb. as the strength of the sling stays having 1-in. bolts, the stays have a factor of safety of only 2.67, and using the highest test figure of 26,650 lb. the factor of safety was found to be 3.26. Calculation shows that the sling stays fitted with 1¼-in. bolts had factors of safety of from 3.67 to 4.15. The tensile strength of the material in the sling stays was shown by test to be 43,200 to 48,300 lb. per square inch and the elongation 18 to 40.5 per cent. in 2-in.. Tests of the sling stays show that the failure was caused by the bolt-holes being drilled too near the ends of the stays. Fig. 6 indicates the manner in which the sling

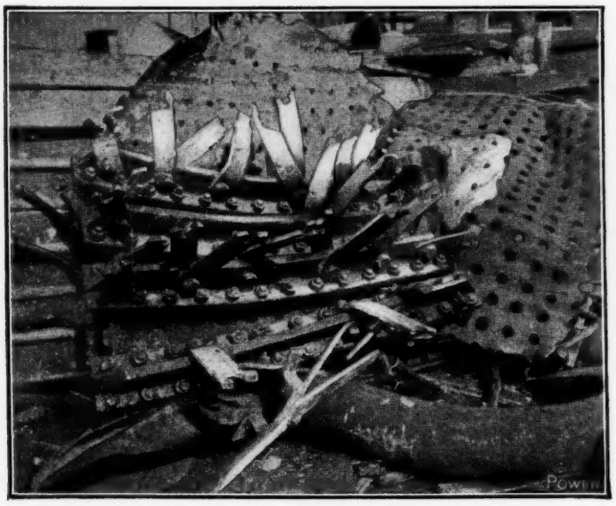


FIG. 4. TOP OF CROWN SHEET, SHOWING HOW SLING STAYS GAVE WAY

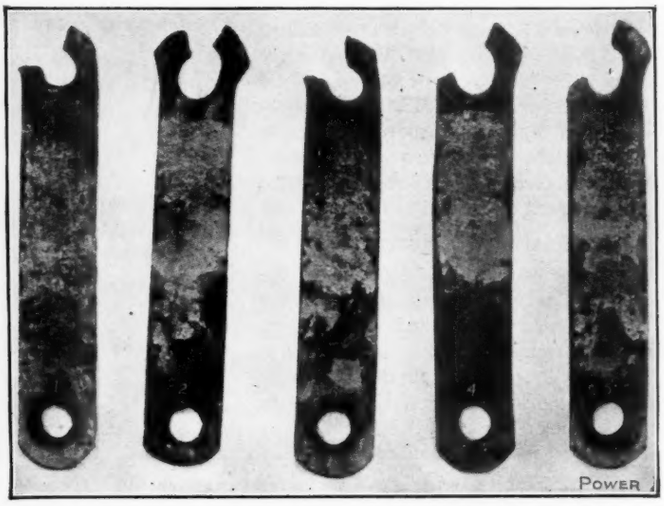


FIG. 5. SHOWING METHOD OF FAILURE AND ELONGATION OF METAL AT EYE OF SLING STAY

stays failed both at the time of the explosion and when under test.

Eighty-six staybolts, nine of which were in the left-side, 63 in the right-side sheet, and 14 in the flue sheet, were found broken at the wrapper sheet and adhering to the firebox sheets. Twenty-six of these staybolts were found to have been partly broken before the explosion. The remaining 60 were in such condition that it cannot be positively stated that they were fractured prior to the accident, but the fact that they broke at the wrapper sheet and did not pull through the firebox sheets indicates a defective condition. Three staybolts taken from this boiler were tested by the Bureau of Standards and the material was found to be of good quality.

Steam Gage Defective

As the steam gage and its connections were destroyed they could not be inspected, but it is probable that the steam gage did not indicate the correct pressure. This could be caused by a defective gage, an obstruction in the siphon pipe, or by a closed or nearly closed valve in the siphon pipe. An inspection of another locomotive of the same class showed that it had two valves in the siphon pipe; when they were open the handle of one formed a right-angle with the pipe and the other was parallel. This arrangement being very confusing and creating a dangerous condition, one valve was ordered removed. It is not known whether such an arrangement of valves existed on the locomotive under discussion.

Inspection Laws Disregarded

The evidence shows that the law and the rules governing the inspection of locomotive boilers were disregarded by the company's inspector and the officials in charge of such work at this point in the matter of making and properly certifying to the reports required by law. A report stating that the safety valves had been set was sworn to on Mar. 16, 1912, by the company's inspector and the roundhouse foreman, who signed it as the officer in charge of such work. The evidence shows conclusively that the safety valves had not been set at that time, and that the explosion occurred while this work was being done. The evidence also shows that the inspector failed to witness the testing of the steam gage and that the injectors had not been tested at the time this report was made out, although he certified under oath that this work had been done.

The flues and firebox sheets show no indication of having been overheated, and the evidence makes it plain that there were three gages of water at the time of the explosion.

Cause of Explosion

The chief inspector and his assistants conclude that this explosion was due to excessive steam pressure caused by a railroad employee tightening the adjusting screw of the safety valves, resulting in an accumulation of steam pressure beyond the endurance of the boiler.

Tests made of the parts of the boiler which evidently failed first demonstrate that the pressure on the boiler at the time of the explosion was greatly in excess of the allowed working pressure. Therefore, the steam gage, either on account of the gage itself being defective or of an obstruction in the siphon pipe between the gage and the boiler, did not correctly indicate the pressure.

The railroad company was at fault in requiring or permitting inspections and reports to be made not in accordance with the law, and in allowing such responsible work as setting safety valves to be performed by an employee of whose experience and judgment, the testimony shows, the management knew practically nothing, and in keeping a boiler in service for which the factor of safety as shown by test was below the recognized standard.

The rules governing the inspection of locomotive boilers, setting of safety valves, testing of gages and similar work are sufficiently comprehensive to insure safety if properly and intelligently complied with.

However, in endeavoring to obviate a recurrence of this accident, necessary action has been taken compelling the use of two steam gages when setting safety valves, one of which must be so connected that it is in full view of the person engaged in setting such valves. Similar action has also been taken requiring the siphon pipe and its connections to the boiler to be cleaned each time the gage is tested.

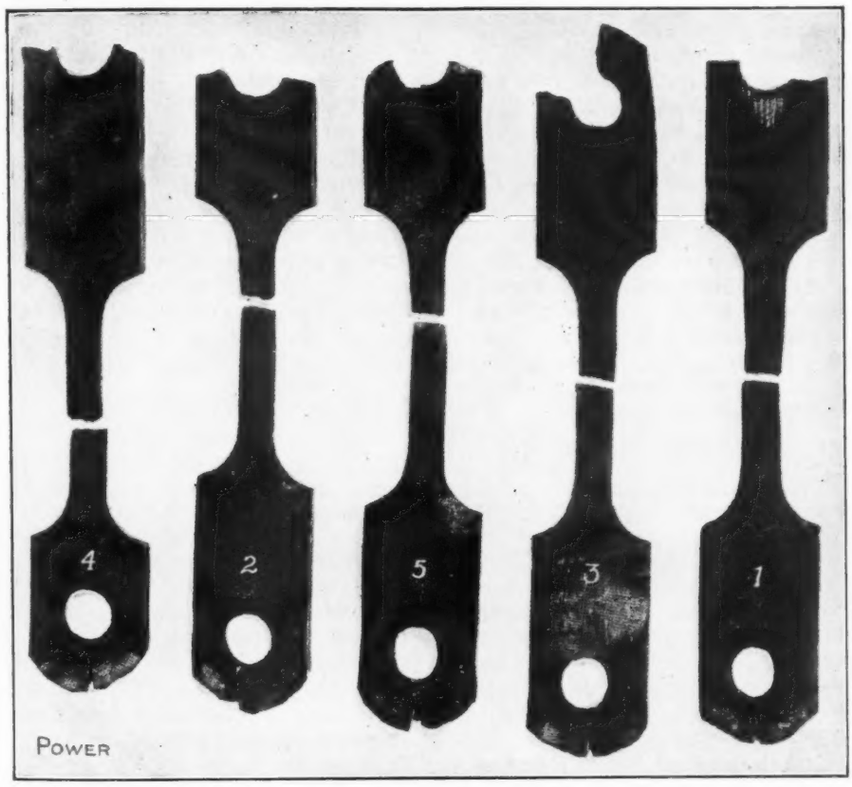


Fig. 6. Results of Tensile Tests on Sling Stays and at Lower End of Tests with 1 and 1¼ inch Bolts

New Mexico's coal production in 1911 was 3,148,158 short tons, with a spot value of $4,525,925, according to a statement by E. W. Parker just made public by the United States Geological Survey. In common with most of the states in the Rocky Mountain region New Mexico produced less coal in 1911 than in 1910, a decrease of 360,163 short tons, or 10.3 per cent. Colfax County, in addition to producing 75 per cent. of the total output of New Mexico, is also the only county in which coking operations are carried on, and in 1911 a total of 767,108 tons of coal mined in that county was made into coke. In 1910 the coal made into coke in Colfax County amounted to 701,204 tons.

The state charter of the Kittanning Water Power Co., a Pittsburg corporation, has been approved by Gov. John K. Tener. The company will give the state entire authority over the construction and operation of its dams and works in Armstrong and Indiana Counties.

Significance of Draft in Boilers

By L. B. Lent

The quantity of gas passing per unit of time is proportional to the pressure drop between two points, the resistance being constant rather than variable.

The above is the title of a bulletin recently published by the U. S. Bureau of Mines and written by Walter T. Ray and Henry Kreisinger. The information it conveys is well worth the serious consideration of anyone interested and will perhaps tend to revise some ideas about this important subject.

Authorities do not seem to agree as to the relation of draft to the amount of combustion or the rate of combustion. Snow says "that for comparison of combustion rates the draft which should be determined is that relating solely to the supply of air to and the overcoming of resistances in the fire. This draft is obviously the difference between the over- and under-grate pressures." This difference between over- and under-grate pressures, Snow calls the effective pressure ("Steam Boiler Practice," page 216).

Kinealy states in his book "Mechanical Draft" that "the number of pounds of coal burned per square foot of grate surface per hour is directly proportional to the square root of the draft." He defines the draft as that measured at the end of the uptake just where it enters the chimney. Evidently the total draft is meant.

According to the Bureau of Mines bulletin, neither of these statements is correct, the rate of combustion being proportional to neither the effective nor the total draft. A clear conception of the meaning of the term "draft" is essential to a proper understanding of what follows. This term is sometimes taken to mean the actual flow of gases, but the proper idea is that it means the difference of pressure which causes the flow. Whether this difference is caused by a chimney (natural draft) or by a fan (induced or forced draft), it is the excess of pressure which pushes the gases through the closed circuit. A flow of gas can never be induced by any "pull" or "suck," as some think. For this reason, the existence of pockets or dead spaces is not the result of any one system more than another, provided the setting is tight.

The movement of gas through the boiler may be compared to that of water through a pipe or to that of electricity through a conductor. The draft producing the flow may be compared to the head of water or to the potential or voltage in the electric circuit. The resistance of the grate, the fuel bed, the gas passages through the furnace, the tubes and other parts determines the amount of air or gas which will flow under the given conditions.

For any given boiler, the resistance of the gas passages from over the fuel bed to the uptake is practically constant; and it is affected in only a slight degree by the condition of the tubes, plates and setting in contact with the gases. The resistance of the fuel bed is apt to be quite variable, depending on the quality of the fuel and the thickness of the bed. Assuming that the latter is the varying resistance and the former the fixed resistance, then this varying resistance may be a greater or less part of the total resistance, according to conditions.

To ascertain the relation of the variable and fixed resistances and of the total resistance, suppose the grate and fuel bed were removed from the boiler. Then this resistance would become zero, the drop in pressure through the grate and fuel bed would be zero and for a given total head the quantity of gas passing would be a maximum. If the grate were replaced without any fuel on it, the slight resistance of the grate would cause a slight pressure drop through it and the quantity of gas passing would, for the same total draft, be slightly less than in the preceding case. If the total draft or pressure difference was 1 in. of water, the drop through the grate might be 0.1 in. of water, the fixed resistance being 0.9 in.

If a thick bed of fuel be placed on the grate, the variable resistance will become very much greater and the quantity of gas passing for the same total head will be very much less.

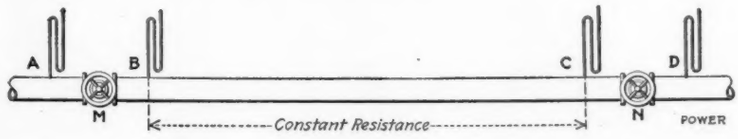
Suppose the grate to be momentarily blanked off so that no gas could pass through it, then the resistance would be a maximum, the pressure above the grate would be the same as that in the uptake and the drop in pressure through the grate would be equal to the total draft. In this limiting case, in which the "effective draft" is greatest, the amount of air flowing is a minimum or zero.

What the writer attempts to show clearly is that a large drop in pressure through the fuel bed (the "effective draft" of some authorities) is simply indicative of a high resistance through the bed and, taken alone, is no indication of the amount of gas passing through the bed. Indeed, a large drop in pressure through the bed indicates a small amount of flow, if anything.

The flow of gas through all parts of the boiler is caused by a diminution of pressure from one part to another and hence a more or less gradual diminution of pressure from the ashpit to the uptake. This drop in pressure from one part to another varies as some power of the resistance to the flow of gases.

1. If the total resistance to the flow of gases remains constant, the pressure drop through any portion of the path of the gases will have a constant ratio to the total drop from the ashpit to the uptake. For example, if the pressure drop through the fuel bed is 0.5 in. of water when the total drop is 1 in., it will be 1 in. if the total drop be increased to 4 in. of water.

2. If the total pressure drop from the ashpit to the uptake remains constant, the pressure drop through any portion of the gas path will vary in the same direction as does the resistance to the flow of gases, although the magnitude of the variations may not be in simple proportion. Suppose the total drop to be 1 in. of water and the drop through the fuel bed is 0.5 in.; now if the total drop be kept the same but the resistance through the fuel bed be doubled by doubling its thickness, the pressure drop through the fuel bed will increase to about 0.65 in. of water; or if for the same total drop, the resistance of the fuel bed be increased by quadrupling its thickness, the drop through the bed will increase to about 0.8 in. of water.



CONSTANT AND VARIABLE RESISTANCE

3. When the resistance through any portion of the gas path remains constant, the weight of gas passing through this portion varies as some power of the pressure drop through this portion. Thus, since the resistance of that portion of the gas from over the fuel bed to the uptake is ordinarily very constant, the weight of gases passing through the boiler varies as some power of the pressure drop between these points.

Approximately only, the weight of gases passing through the boiler is directly proportional to the square root of the pressure drop through the boiler proper or

through any other portion of the gas path having a constant resistance.

It should be evident from the foregoing that while it necessarily requires a comparatively large drop in pressure to drive air through a thick bed of fuel, this large drop is no measure of the amount of air flowing, and since the rate of combustion depends on the amount of air supplied per unit of time, the drop in pressure through the fuel bed (the "effective draft") is no measure of the rate of combustion.

As previously pointed out, the better measure of the rate of combustion is the pressure drop between the upper side of the grate and the uptake, or any two other points between which the resistance is constant.

Without carefully considering the matter, the inclination is to think that a large difference in pressure over and under the fuel bed must necessarily induce a correspondingly large flow and hence the effective draft through the fire, when large, must produce a large flow of air and a corresponding high rate of combustion. It is forgotten that the real reason for this large difference of pressure is the existence of a high resistance to the flow of gases and that the total head is the available one, the one which really limits the rate of combustion.

Several charts shown in the bulletin and plotted from the test results could be introduced here, but it is sufficient to say that the results obtained from horizontal tubular, water-tube and locomotive boilers show a very close substantiation of the foregoing statements.

Consider for a moment the straight piece of water pipe arranged as shown in the accompanying illustration with two valves and four pressure gages. The straight length of pipe between *B* and *C* has a constant resistance due to its friction alone; the valves constitute variable resistances, depending on how far they are open. Supposing the valve *M* is opened and *N* is closed, the gage *C* will show a 10-in. pressure and no flow will take place. If the valve *N* is cracked open, a small flow will take place and the pressure *C* will be slightly less than *AB*. To be exact, the difference in pressure between *B* and *C* will be the amount necessary to induce a flow at the resulting velocity against the fixed resistance of the pipe friction. The greater the difference in pressure, the greater will be the velocity and the weight of water passing.

With a pressure of 10 in. of water at *A*, the pressure at *B* will depend on how much the valve *M* is opened. Suppose the valve *N* to be fully open so that the flow is unrestricted. Then, if the valve *M* is cracked open, the pressure at *B* will be perhaps 1 lb., the friction of the partially opened valve *M* absorbing the other 9 in. of pressure. The flow of water through the pipe will be that due to the small difference of head between *B* and *C*. As the valve *M* is gradually opened, the pressure *B* approaches that at *A*, the difference in pressure between *B* and *C* increases and the flow of water increases proportionately.

It is a well known law of hydraulics that if the difference in head and the resistance are known, the velocity can be calculated; and for a given size pipe, the volume is proportional to the velocity. If the resistance between *A* and *B* through the valve *M* were known, the velocity could be calculated. But the resistance of the valve *M* is variable and depends on the amount it is open; whereas the resistance of the section of pipe between *B* and *C* is constant and can be measured very accurately from the drop in pressure between those two points.

In any given case, the amount of water passing through the valve *M* in a given interval of time is exactly equal to that passing any other point in the line in the same interval of time. This is one of the basic laws of hydraulics and is sometimes spoken of as the principle of continuity of flow, the law being that the quantity of water passing any cross-section of the passage, no matter what its shape or velocity at the cross-sectional area, is exactly the same as that which passes every other cross-section. This is another way of saying that the same quantity of water comes out one end of a pipe as goes in the other end in the same time.

Assume that the valve *M* may be replaced by the variable resistance of the fuel bed in the boiler and the section of pipe between *B* and *C* by the fixed resistance between the upper side of the fuel bed and the uptake, and practically the same laws hold. As the resistance at *M* increases (by making the fuel bed thicker) the effective head at *B* decreases, the difference in head between *B* and *C* decreases and a less quantity of gas flows. What should be emphasized is the fact that, although the pressures at *A* and *B* may be known, unless the numerical value of the resistance through the valve *M* is known, the quantity of gas passing cannot be calculated nor indicated.

Since the resistance of the gas path between the upper side of the grate and the uptake is practically constant, the quantity of gas passing is proportional to the difference in pressure between these two points; since the same quantity passes through the grate per unit of time as passes these points, the difference in pressure between these points is proportional to the amount of gas passing through the fuel bed and is hence proportional to the rate of combustion.

As the authors of this bulletin carefully point out, the quantity of gas passing is not directly proportional to the difference in pressures, but to some power of that difference. The important point is that the quantity of gas passing per unit of time is proportional to the pressure drop between two points, between which the resistance is constant, rather than to the pressure drop between two points, the resistance between which is variable.

Those desiring to read the full text of the bulletin from which these facts and ideas are taken may obtain it from the Bureau of Mines, Washington, D. C.

Rule for Horsepower

By F. R. Low

A rule for figuring the horsepower of an engine, especially convenient for use with the slide rule, is

Multiply the square of the diameter by 238, *by the piston speed and by the mean effective pressure.*

Point off seven decimal places more than there are decimal places in the factors used.

The 238 of the rule is the quotient of 7854 by 33.

$$Hp. = \frac{Area \times piston\ speed \times m.e.p.}{33{,}000}$$

$$Area = diameter^2 \times 0.7854$$

$$Hp. = \frac{diam.^2 \times 0.7854 \times piston\ speed \times m.e.p.}{33{,}000}$$

The 33,000 divides the 0.7854 evenly, the quotient being 0.0000238; so that

$$Hp. = D^2 \times 0.0000238 \times p.s. \times m.e.p.$$

Example—What horsepower is developed by a 12x24-in. engine, running at 130 r.p.m. with a mean effective pressure of 42 lb.?

$$12 \times 12 = 144 = D^2$$
$$238 = \text{constant}$$
$$1152$$
$$432$$
$$288$$
$$\frac{24 \times 2 \times 130}{12} = 520 = \text{piston speed}$$
$$685440$$
$$171360$$
$$17821440$$
$$42 = \text{m.e.p.}$$
$$35642880$$
$$71285760$$
$$74.8500480 = \text{hp.}$$

Seven decimal places are pointed off in the final product because there were no decimal places in the other figures. There will be no difficulty about this pointing off. It is self-evident that a 12-in. engine is doing more than 7.48, and not so much as 748 hp., so that one will never stop to count decimal places. A rough index of the capacity of a normal engine is one-half the square of the diameter, in this case 72.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Catechism of Electricity

SUBSTATIONS

1277. *What is a substation?*

An electrical station in which current is received from one or more large generating stations and in turn supplies a number of feeders which distribute it over the locality where it is used.

1278. *Under what conditions is a substation mostly used?*

In electric-railway work, where the traffic is heavy and the system of distribution widespread; although in some cases it is used for power and lighting systems.

1279. *What advantage is gained by the use of a substation in electric-railway work?*

One large power station can be used for the entire system. This enables the electrical energy to be more economically obtained than when more generating stations are employed.

1280. *Is the current handled in a railway substation direct or alternating?*

The currents received at the substation from the generating plant are alternating and of high voltage. These are stepped down by transformers, and by means of rotary converters are changed into direct curent, usually at 550 volts, and supplied to the feeders.

1281. *Why are transformers required in the substation?*

To lower the voltage of the alternating currents supplied from the generating station, so that they are adapted to the rotary converters. This is necessary on account of the fixed ratio which the voltage on the alternating-current side of a rotary converter bears to that on the direct-current side. In a three-phase converter this ratio is 61 per cent.; hence alternating current at approximately 335 volts would be necessary to supply direct current at 550 volts.

1282. *Are motor generators used in substation work?*

Motor generators are seldom used on account of their higher cost as compared with the rotary converters and transformers they would replace.

1283. *Are all substations alike with respect to the method of receiving and distributing the power?*

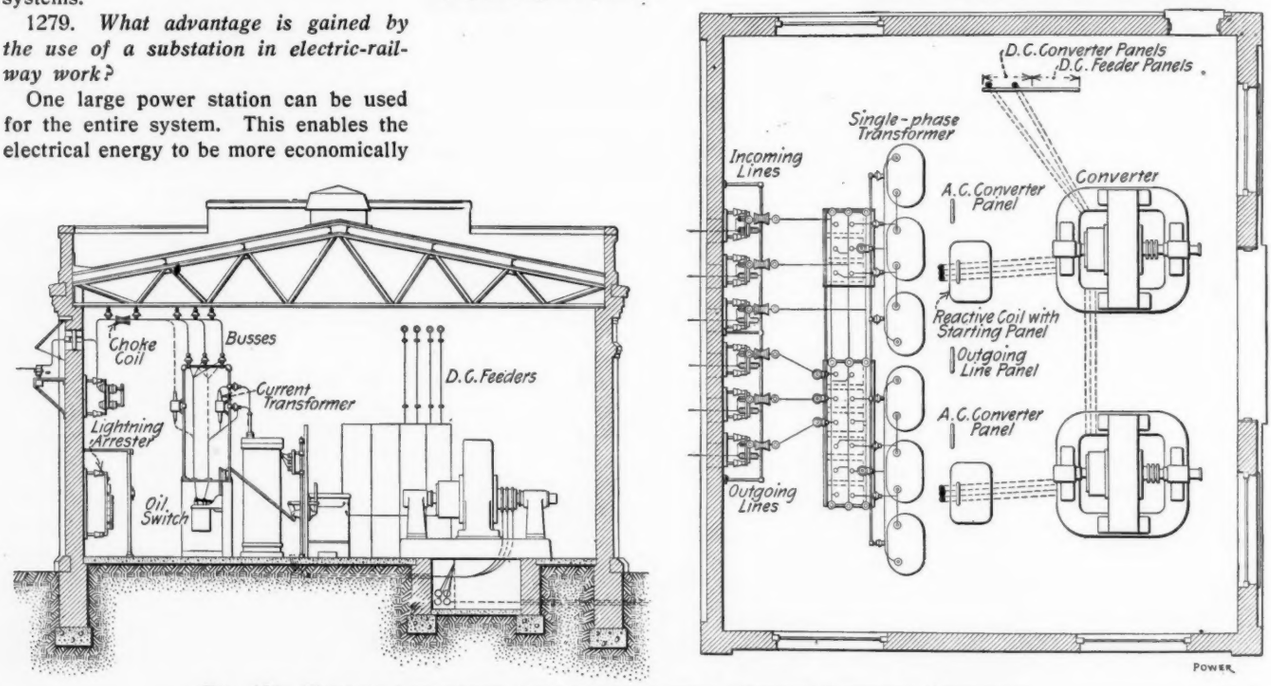


FIG. 421. TYPICAL SUBSTATION ARRANGEMENT WITH FEEDERS ENTERING OVERHEAD

No. A substation may receive current from the generating station and without any transformations whatever distribute it through switches to other circuits. Another kind of substation may receive high-voltage alternating current from the generating station and, through transformers, reduce its voltage for distribution. A third, as previously described, may receive high-voltage alternating current and distribute low-voltage direct current. Still other substations, common in electric-railway service, carry either storage batteries alone or storage batteries and boosters.

1284. *When are storage-battery substations advantageous for railway work?*

When maintaining a more uniform line voltage and compensating for an extra length of feed wire; relieving the power house of fluctuations of load, keeping the cars moving when the power supply is temporarily interrupted, or in supplying power for the operation of a few cars or lights when the power house is shut down.

1285. *When are boosters used in connection with a storage battery in a substation?*

When current is to be transmitted over a considerable distance to a storage-battery substation for the purposes mentioned in 1284 and the battery must furnish current at the same voltage as developed in the generating plant. The booster makes up for the drop of potential in the line between the generating station and the substation, while the battery is being charged. Also, in railway work, where the load fluctuates within wide limits, a booster is used in connection with the battery. In this case a "differential booster" is used having both series and shunt field windings. Thus the e.m.f. varies automatically with the load and adds to or subtracts from the battery voltage as required.

1286. *What factors govern the location of a substation?*

Practically the same factors that govern the location of a central station, namely: nearness to the center of distribution so

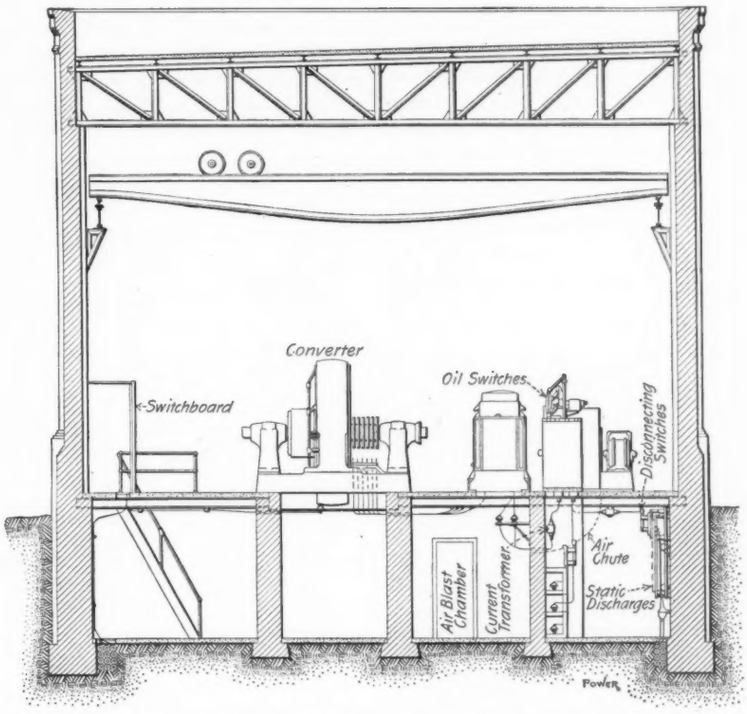


FIG. 422. SECTION THROUGH SUBSTATION WITH UNDERGROUND CABLES

that the drop of potential in the different circuits may be a minimum, also the cost of real estate.

1287. *What kind of a building is best for a substation?*

A detached building, built preferably of brick or other fireproof material.

1288. *What provision should be made for light?*

The substation should be constructed so that plenty of light will enter it, but in a storage-battery room the sun should not shine on any part of the battery. A substation used exclusively for a storage battery is sometimes built without windows, dependence being placed entirely upon incandescent electric lamps for light.

1289. *What provision should be made for ventilation?*

The ventilation of the storage-battery room must be particularly good to prevent the accumulation of fumes; otherwise, it will be almost impossible to enter the room while the battery is being charged, and the hydrogen gas given off while charging, being imflammable, is likely to cause a serious accident if ignited.

The room in which the machines are installed should also have good ventilation because the heat developed in them, even during their normal operation, will considerably increase the temperature. Good ventilation is likewise important where oil-cooled transformers are used, so that the temperature may not become unduly high in the summer-time.

1290. *Is there any reason why the machines should not be located in the battery room?*

The battery should be placed in a room by itself so that the sulphuric-acid fumes given off from the cells will not corrode the metal of the machines.

1291. *What provision should be made for the drainage of a substation?*

Where air-blast transformers are used, the walls of the air-blast chamber should be waterproofed and the substation should be built at such an elevation that water will not stand on the floor of the air-blast chamber. If this latter precaution is not taken, the transformers may be damaged by the warm air from the blower picking up moisture and depositing it in the transformers which are not in service.

When oil-cooled transformers are used, it is well to install a pit of sufficient capacity to drain the oil from several transformers, and to provide drainage piping of ample size from the oil drain-cocks on the transformers to the pit, so that the oil can be drawn off quickly in case of fire or other emergency.

In the battery room the floor, which must never be of wood but of vitrified brick or some other material that is not affected by sulphuric acid, should slope downward toward a drain so that when flooded in washing, it will dry off rapidly.

Where cables come into the station underground, the entering conduits should be sealed and suitable drainage provided so that water cannot leak into the cellar through these openings.

1292. *Give an idea of the general arrangement of substation apparatus.*

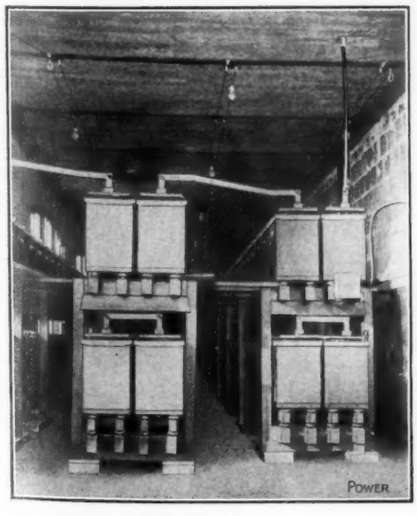


FIG. 423. ARRANGEMENT OF STORAGE BATTERIES WHERE FLOOR SPACE IS LIMITED

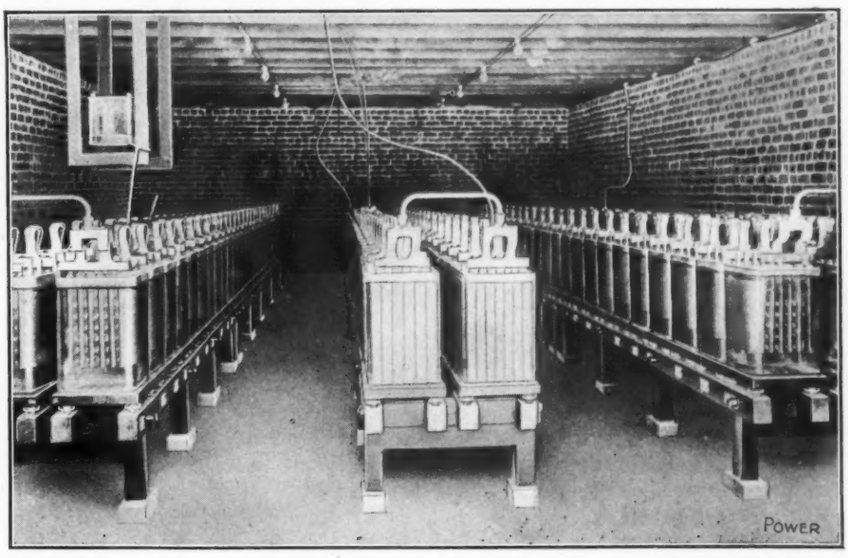


FIG. 424. BATTERIES WHERE FLOOR SPACE IS NOT LIMITED

The arrangement of substation apparatus is largely dependent upon the nature of the switching apparatus. When the cables enter the station overhead, the hand-operated switches are invariably of the top-connected type and the high-voltage busbars are mounted above them. A typical arrangement of this kind is shown in elevation in Fig. 421, which

FIG. 425. PORTABLE RAILWAY SUBSTATION

represents an 11,000-volt installation where oil-insulated self-cooling transformers are used. The high-tension panels are located in front of their respective oil switches and on the operating side of the transformers.

1293. *What arrangement of substation apparatus is customary in case the incoming cables are underground and the switches are motor-operated?*

Fig. 422 shows a section through a substation to meet these conditions. The switches are bottom-connected and the busbars and all high-tension connections are placed below the floor in a suitable compartment. The connections from the high-tension switches to the transformers pass through the partition wall between the high-tension compartment in the basement and the air-blast chamber. The transformers in this case are bottom-connected; that is, both the high-tension and the low-tension leads are brought through their bases. All the main panels, both alternating-current and direct-current, are in one switchboard.

1294. *What is the usual arrangement for the switchboard panels?*

In general, the direct-current switchboard panels are arranged in a group by themselves, the converter panels to the left and the feeder panels to the right, with room for extension at either end. Usually no attempt is made to group the oil switches, either hand-operated or electrically operated, behind the main switchboard, the oil switches being invariably located, in the more recent types of construction, immediately adjacent to their banks of transformers or line entrances, or exits, as the case may be. For all stations using hand-operated switches, this makes it advisable to locate the alternating-current line and converter panels in corresponding positions. With electrically operated switches, there is no definite relation between the location of the panel and its switches. No incoming line panels are used unless the lines are in duplicate, a single incoming line being wired to the substation busbars without oil switches, but with disconnecting switches.

1295. *Upon what considerations does the arrangement of a storage-battery substation depend?*

Principally upon the amount of available floor space and the accessibility of the cells for inspection and handling.

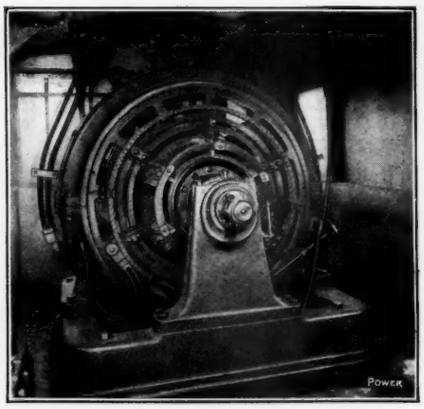


FIG. 426. CONVERTER IN PORTABLE SUBSTATION

1296. *When the floor space is limited, what is the best arrangement of the cells?*

In tiers, one above the other, as shown in Fig. 423.

1297. *Are the cells in Fig. 423 convenient for inspection and handling?*

Inspection of the cells is not difficult in this installation because ample room has been left between the two tiers and between the ceiling and the upper tier. The handling of the cells, however, is not convenient; without a traveling crane it would be almost impossible to move any of these cells with ease and safety.

1298. *Where the floor space is not limited, what is the best arrangement of the cells?*

In rows on supports resting upon the floor as in Fig. 424.

1299. *Is there any objection to placing the cells directly upon the floor?*

It is not advisable to do this because moisture would accumulate under the cells, and this, together with the battery solution, would form a conducting path from one cell to another and cause leakage of current. Hence, it is customary to use insulating supports under the cells.

1300. *What is gained by having the cells rest upon a board over the insulators as in Fig. 424, rather than directly upon the insulators?*

The board prevents any unequal distribution of the weight of the battery from causing the insulators to be strained or broken.

1301. *Is there any objection, where the space is limited, to placing adjacent cells so close to each other that their cases touch?*

Adjacent cells must be kept far enough apart so that there will be no possibility of coming in contact, else the battery solution is likely to creep across, causing leakage of current.

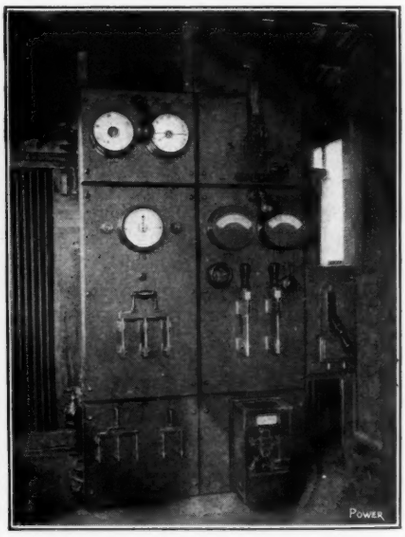


FIG. 427. SWITCHBOARD IN PORTABLE SUBSTATION

1302. *Are provisions ever made in a railroad substation, other than by the overload capacity of the apparatus installed, for taking care of unusually heavy loads at infrequent intervals?*

A portable substation is sometimes used for this purpose.

1303. *What is a portable substation?*

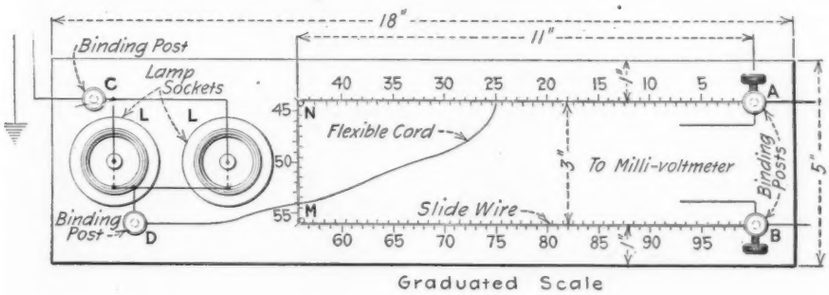
A car such as shown in Fig. 425 arranged for carrying a rotary converter (Fig. 426), transformers, and switchboard (Fig. 427). Such a station for increasing the direct-current supply may be quickly moved to any part of the line, as to fair grounds, amusement parks or summer resorts which are to be served only a few days or weeks.

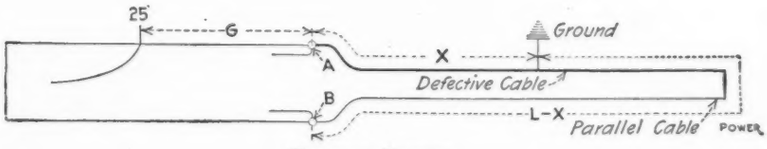
Locating Cable Grounds

By J. W. Himmelsbach

In underground distributing systems, one or more conductors may become grounded, and circumstances demand that that particular conductor be back in service quickly. Unless the ground produces such violent disturbances that no doubt exists of its location, it is necessary to either cut until the fault is located, which is generally very slow, or else subject the grounded conductor to tests to determine the exact location.

The very satisfactory apparatus described can be easily and cheaply made. The only expensive part is a direct-reading, differential millivoltmeter having the zero in the middle of the scale, which reads 75 millivolts on either side of the zero point. Mount on a piece of well seasoned 5x18x1-in. oak four binding-posts, *A*, *B*, *C*, *D*, and two lamp sockets, *L*, *L*, as shown; *M* and *N* are two small wire brads driven in the board, leaving only ⅛ in. projecting.





Testing Outfit

From *A* to *N*, *N* to *M* and *M* to *B*, paste strips of paper 1 in. wide upon which a scale has been drawn, with divisions every ⅛ in., which will give 200 divisions. Mark every second division from 0 to 100, starting at *A*. Then stretch a piece of No. 26 or 28 B. & S. gage, german-silver wire from *A* around *M* and *N* to *B*. This wire must be stretched tightly so that reading the scale will give correct proportional lengths of wire.

The lamp sockets are to be wired in parallel. Connect permanently to *D* a piece of flexible wire (lamp cord will do) to the end of which is soldered a knife-edge contact. This wire must reach from post *D* to post *A*. Binding posts *A* and *B* must have two connectors, as two sets of leads are fastened to them; this completes the testing set.

To prepare for the test, connect to the binding posts *A* and *B* respectively, one lead to the station end of the grounded conductor and one lead to a conductor parallel to the grounded conductor and having the same destination. The ends of these two conductors, away from the testing set, must be joined together. Connect the millivoltmeter to the posts *A* and *B*.

If a 125-volt direct-current circuit is convenient, connect one side to post *C* and ground the other, and place two 16-cp. lamps in the sockets. If no direct-current circuit is available, five or six battery cells connected in series can be used. Connect one terminal to *C* and ground the other; short-circuit the lamp sockets with a plug fuse.

To locate the grounded point, run the knife-edge contact connected to *D* along the graduated wire until the millivoltmeter reads zero. Suppose the reading on the wire is 25 divisions from *A*. Referring to the lower diagram, if the total length of the cable loop is *L*, and the distance from the station to the ground is *X*, then the following proportion holds good:

$$25 : X = 75 : L - X$$

solving for *X*,

$$75\,X = 25\,L - 25\,X$$

$$100\,X = 25\,L$$

$$X = 0.25\,L$$

and if the length of the cable is known, the distance *X* can readily be determined.

Calling the distance from *A* to the point on the slide wire which gives zero deflection on the millivoltmeter *G*, and the distance from *B* around to this point *H*, also the total loop length of the conductor, and the distance from the station to the ground, *L* and *X*, respectively, as before, then:

$$G : X = H : L - X$$

$$G\,L - G\,X = H\,X \qquad (1)$$

but *G* plus *H* equals 100; therefore,

$$H = 100 - G \qquad (2)$$

substituting (2) in (1),

$$G\,L - G\,X = 100\,X - G\,X$$

$$100\,X = G\,L$$

$$X = \frac{G \times L}{100}$$

which is the formula to be used when locating grounds with this apparatus.

If the ground is due to water—which will mean that it is not confined to one point—this method is not very satisfactory. If two or three conductors in the faulty cable are grounded, thus making it impossible to get a cable clear from ground for a return, it will, in all probability, be unnecessary to make the location test as a double ground is equivalent to a short-circuit, and short-circuits are usually very apparent.

LETTERS

Commutator Lubrication

I have read with interest Mr. Barrick's letter in the Apr. 30 issue on "Commutator Lubrication." I have had charge of direct current machines for years and do not use any oil or commutator compound whatever, as such things, in my experience, have led to trouble, causing dirt to collect on the comutator and bushes. If Mr. Barrick's commutators are turned off properly, the brushes set correctly and all the connections kept clean, he will have no trouble with sparking provided the machines are properly designed and the load kept within reasonable limits. The use of sandpaper should be discouraged as it soon causes the commutator to be out of true, and also leads to high mica.

One machine in my charge was turned off last in May, 1910, and after carrying 10 per cent. overload during half that time, it has not been turned since.

George Drewry.

Brantford, Ont.

Mr. Barrick's letter recalls to me the method used in a power plant where I was recently employed. A piece of rod packing was kept handy and applied to the surface of the commutators several times a day. This method not only lubricated the commutator but also cleaned them. Commutators treated in this way will keep very smooth, neither sparking nor showing any wear. My chief objection to using oil on a commutator is that it collects on the brushes and makes a sticky base to collect all sorts of dirt.

Frederick Hansa.

New York City.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Trebert Gasoline Engine

One of the most interesting exhibits at the recent aëroplane show in New York was the Trebert engine, a six-cylinder four-stroke-cycle machine in which the cylinders rotate about a stationary shaft. However, unlike the few other engines built on this principle, such as the "Gnome," the cylinders are placed around the shaft with their axes parallel instead of radial to it.

As may be seen from the illustrations the cylinders, which are cast separately, are held in position at the rear by an aluminum crank case and at the front by a ring containing the individual cylinder ports. No valves are used for exhaust or intake, these events being controlled by a stationary disk chamber surrounded by a rotating ring which uncovers the ports at the proper time. The arrangement is as follows:

Attached to the axial shaft is a stationary disk-shaped gas and exhaust chamber divided into two separate compartments with air spaces between, although the outer cylindrical surface is continuous. This also constitutes part of the front bearing for the rotating parts. A pipe leads from the carburetor to the intake compartment, and the exhaust compartment is connected to a muffler, when desired. Around this gas and exhaust chamber fits a revolvable ring carrying the individual cylinder port chambers, each of which is fitted with a packing sleeve bearing lightly against the cylindrical face of the stationary chamber and held in place by springs which counteract any centrifugal force. Holes are bored in the heads of these sleeves to permit an equalization of pressure and thus avoid jamming the springs during compression. In addition, compression plates are fitted between the stationary and the revolving parts to prevent leakage of the gas.

In the cylindrical face of the stationary disk are two elliptical ports, one from the intake chamber and the other lead-

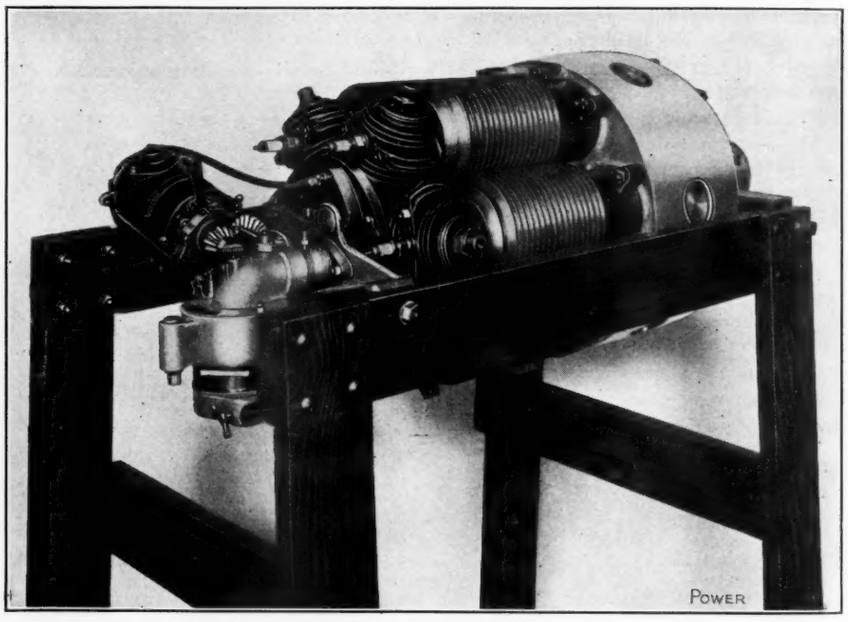


FIG. 1. VIEW OF ENGINE FROM FRONT

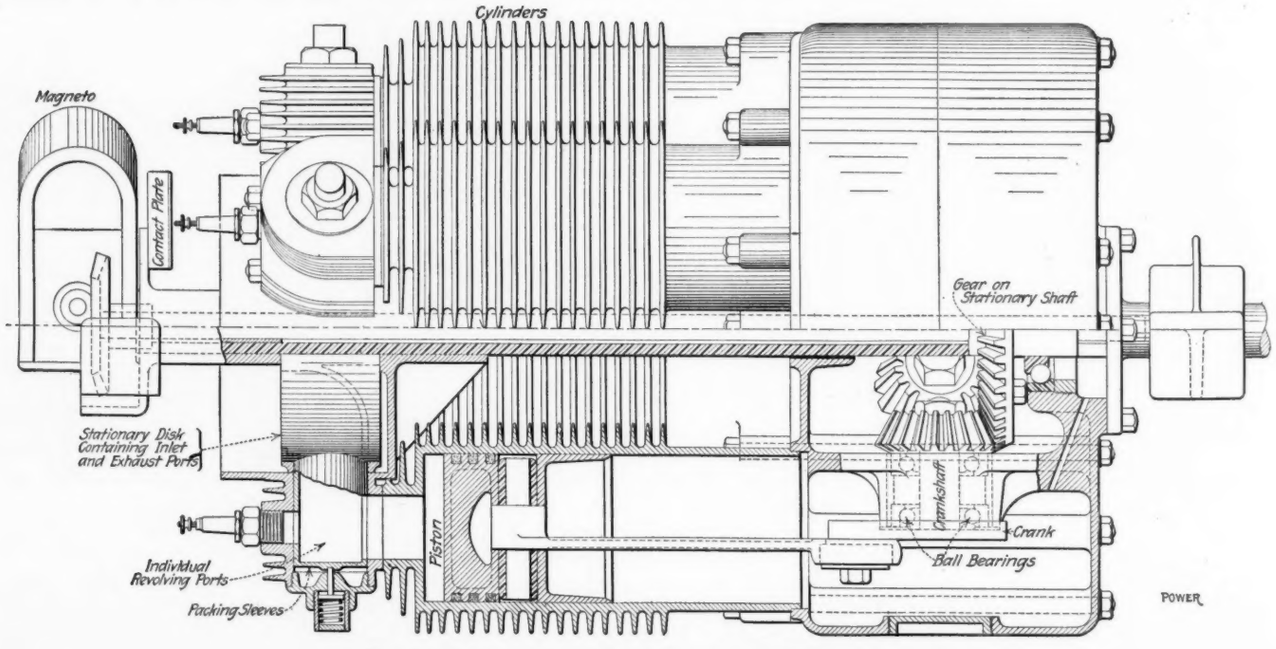


FIG. 2. ELEVATION AND HALF-SECTION OF TREBERT ENGINE

ing to the exhaust pipe, spaced the required angle apart so that the cylinder is always ready to receive gas or to exhaust when its sleeve slides over the proper port. The fresh mixture and burnt gases cannot mingle.

The ignition is extremely simple. The magneto, which is placed at the front of the engine, is driven through a shaft and bevel gear at three times the engine speed. A spark plug is attached to each cylinder-port chamber and in the course of one revolution each passes over the contact plate connected with the magneto. Thus a spark occurs in each cylinder every revolution, and it may be advanced or retarded by a lever which shifts the position of the contact plate. Thus the engine is practically its own timer and distributor.

In the crank case are six crankshafts spaced radially 60 deg. apart—one for each cylinder. They run in ball bearings supported in ribs integral with the crank case, and their inner ends carry bevel gears which mesh with a large bevel gear on the stationary shaft. The outer ends carry cranks which are driven through connecting-rods by the pistons. Thus the whole motor, except the gas and exhaust chamber, and the carburetor and magneto, revolves about the axial shaft which is secured to the front suspension member. The driving shaft projects from the rear of the crank case.

The lubrication also is very simple, three pipes with adjustable sight-feeds sufficing for the whole engine. The cylinders and pistons are lubricated through the gas intake, the oil being introduced into the intake pipe after the gas has left the carburetor. The crankshafts, driving gears and bearings are oiled by two lines running through the center of the axial shaft and emerging just in front of the large gear. From this point the oil is distributed by centrifugal force.

The engine shown in the illustrations is rated at 60 hp. when running at 1000 r.p.m. The cylinders are 3¼-in. bore by 4-in. stroke and the whole engine occupies a space of 32x16x16 in. The weight complete is about 250 lb. The rotating parts themselves provide the necessary flywheel effect, dispensing with the use of a flywheel.

There being six cylinders, each is required to develop only 10 hp. and a complete cycle occurs every revolution (as in a two-stroke-cycle engine); hence there are 6000 power strokes per minute.

While the circulation of air produced by the rotating parts is sufficient to keep the cylinders from becoming overheated with this size of engine, in the larger sizes it has been found advisable to water-cool the cylinders.

CORRESPONDENCE

Backfiring Overcome

As a gas-engine repairman it fell to my lot to work on a small engine installed in a printing plant; it was giving a great deal of trouble by backfiring. The engine was of the four-stroke-cycle, two-cylinder opposed type and was to run on illuminating gas from the city mains. The inlet valves were automatic (operated by the suction of the piston) and both cylinders drew gas from the same mixing valve.

The ignition equipment comprised a small dynamo and a set of batteries for starting, both of which furnished current to a single induction coil. Both primary terminals were led to binding posts, one on the case and the other on the timing device, and each secondary terminal was connected to a spark plug.

The timing device consisted simply of contact pieces *C* (Fig. 1) mounted, one on each exhaust-valve liftrod, which were forced by springs against other stationary contacts *D* mounted on but insulated from the engine frame. One side of the exhaust cam was flattened to allow the roller follower (not shown) to drop and thus make the contact at the proper time. The stationary contacts were insulated from the engine frame, but not from each other.

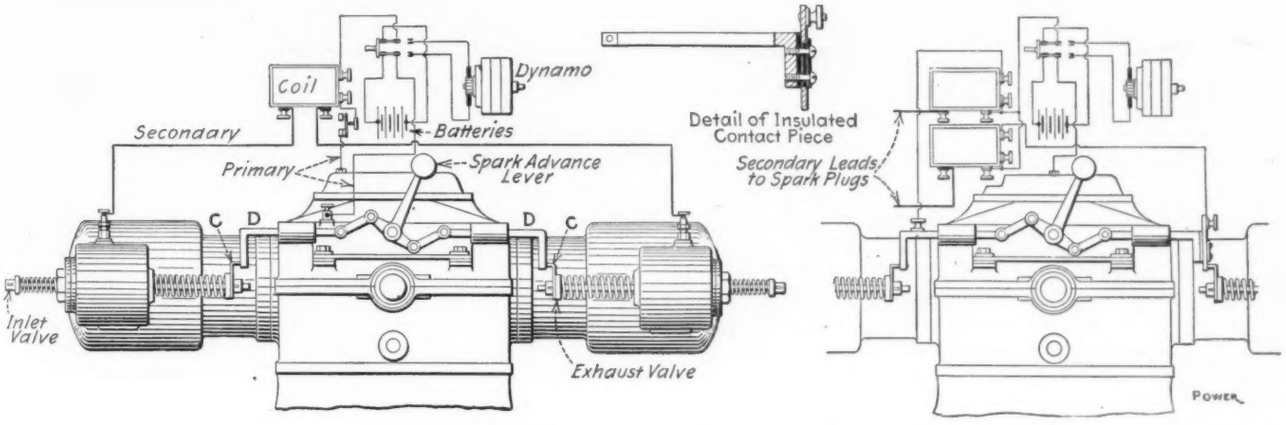
This arrangement caused a spark to occur in both cylinders at the same time. As both pistons travel toward the inner center (away from the cylinder head) at the same time, one on the suction and the other on the power stroke, it is plain that the contact must be broken before the pistons are very far past this center; otherwise the incoming gas of the cylinder which is on its suction stroke will be ignited and cause backfiring.

In this case, the uneven quality of the gas and the variable pressure made a long contact necessary in order to properly ignite the charge, and advancing the spark to the running point would not give an early enough break to eliminate the trouble.

The difficulty was finally remedied by removing one of the stationary contact pieces *C* and replacing it with one which was insulated from both the engine frame and the other contact piece. Another spark coil was added (see Fig. 2), giving one coil for each cylinder, and the trouble so far as backfiring was concerned was eliminated.

J. Arthur Fish.

Kidders, N. Y.



Original Ignition System and that After Altering

The new artificial gas plant installed in East Liverpool, Ohio, is attracting much attention because of the work being done. The builders claimed that from one ton of coal they could produce 40,000 cu.ft. of gas, and the tests so far have shown that they are getting from 55,000 to 70,000 ft. The plant was built to run from 10,000,000 to 12,000,000 cu.ft. a month, but it is expected that the machinery will be able to produce a far greater amount.

For the year 1910, reports the Department of Mines of Western Australia, there were 58 gas engines in use in mines developing 607 hp., while 2933 hp. was developed by 420 oil engines.

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Precooling Plant of Southern Pacific

By LeRoy W. Allison

The Southern Pacific Ry. Co. has erected a combined ice-manufacturing and precooling plant at Colton, Calif., to assist primarily in the transportation of perishable fruits, vegetables, etc., from the agricultural belt of southern California. It is one of the largest of such plants, having a capacity of 40 loaded cars, and is operated by the Pacific Fruit Express Co., a subsidiary organization.

The plant has been in active service since the summer of 1909 and embodies the direct-expansion system of refrigeration and a precooling process known as the intermittent vacuum system, offering distinctive and interesting features. The entire plant was designed by A. Faget, of San Francisco; the patents on the precooling process are controlled by L. H. Roy, Chicago, Ill.

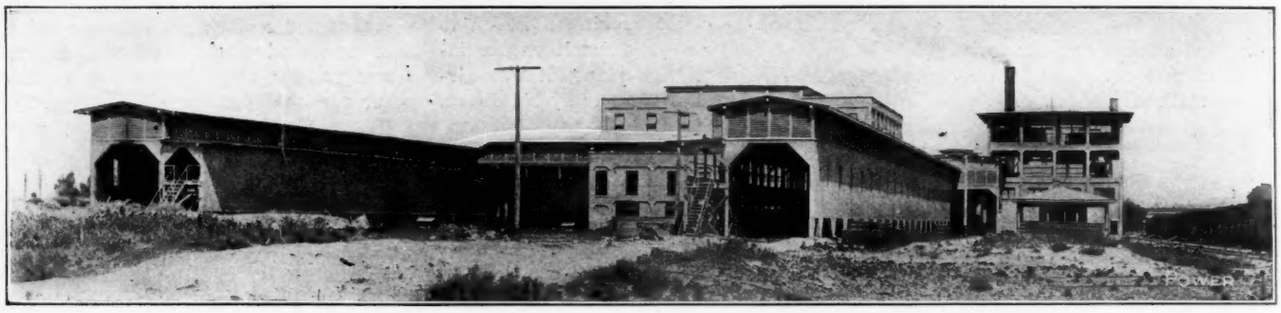


Fig. 1. The S. P. R. Precooling Plant, at Colton, Calif.

General Plan

The entire plant, Fig. 1, excepting the car shed, is built of reinforced concrete. The building at the extreme right of the illustration is the power plant and comprises two 225-ton York refrigerating machines of the double-acting type with ammonia compressors 24x40 in., driven by a tandem-compound Corliss engine 24 and 46 by 36-in. stroke. Steam is supplied by three 250-hp. Stirling boilers, set two in a battery and one single. The usual power-plant auxiliaries, boiler-feed pump, oil-feed pump, circulating pump, fire pump, etc., are installed.

The condensing apparatus both for the steam and ammonia systems are installed in the overhead open portion of the power plant, as may be noticed in Fig. 1. The former installation, on the second floor, is composed of Wheeler condensers. The ammonia system, on the top floor, consists of four sets of ammonia condensers 20 ft. long and 24 pipes high, having 48 coils of 2-in. galvanized pipe. Water for this system, as well as for ice manufacture, is obtained from neighboring artesian wells and irrigation ditches connected to a concrete reservoir at the rear of the power plant. This reservoir is at the surface of the ground and is approximately 8 and 20 by 7 ft. deep.

Electric energy for operation is purchased from a local central station; it is delivered at 11,000 volts and reduced at the company's individual transformer station, adjacent to the power plant, to 220 volts for motor and kindred service.

The main building, centrally located between the two train sheds and icing docks, is 111 ft. 2 in. wide by 276 ft. 8 in. long, and contains the ice-manufacturing and storage plants. The freezing tanks have a capacity of 250 tons daily in blocks of 300 lb. each; the plant is fully equipped with modern ice-making auxiliary equipment and measuring apparatus. Beneath the ice plant a day storage chamber with capacity of 300 tons every 24 hr. is built, while two adjacent winter storage rooms, with total capacity of 17,500 tons, are similarly constructed. The insulation in the building consists of lith board and cork, supplied and installed by the Union Fiber Co., of Winona, Minn. In the storage rooms, this waterproof lith board is 3 in. thick and is

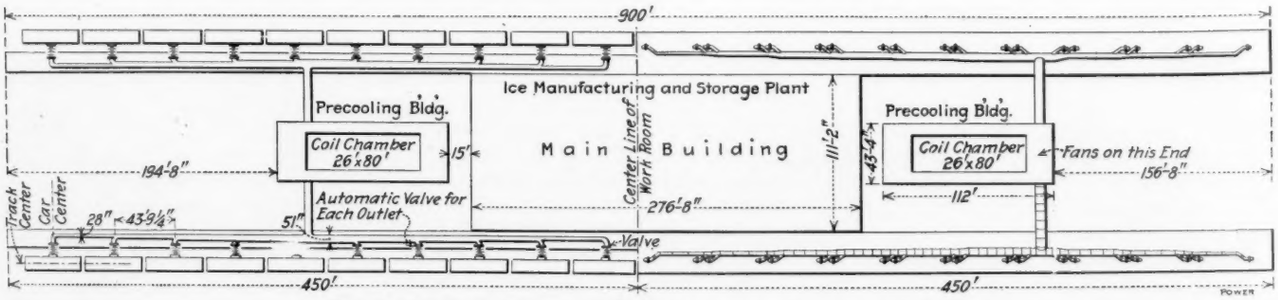


Fig. 2. General Layout of Precooling and Ice-storage Plant

bedded in a thick coating of cement; on the outside it is faced with corkboard and waterproof cement.

Precooler Coil Chambers

As may be noticed in Fig. 2, two precooling plants are located at either end of the ice-manufacturing plant. They are each 43 ft. 4 in. wide by 121 ft. long and identical in construction, so only one need be considered. The main portion, as seen in the forepart of Fig. 1, is

two stories high, the central portion, directly over the coil chamber, having a half-third story or ventilating deck. An interior view of this section of the building is shown in Fig. 3. The coil chamber is 26x80 ft., and comprises upper and lower sections. The walls are constructed of hollow terra-cotta tile, a double row each 4 in. thick, with two layers of 2-in. lith board insulation between. On the outside, 4-in. corkboard with a pitch asphaltum covering is laid. The ceiling is of the same type of construction, with corkboard on the exterior, while on the floor the insulation is laid on 5 in. of concrete and covered with 2 in. of finished cement concrete. The upper and lower chambers are identical and are used for exhaust from and inlet to the car, respectively.

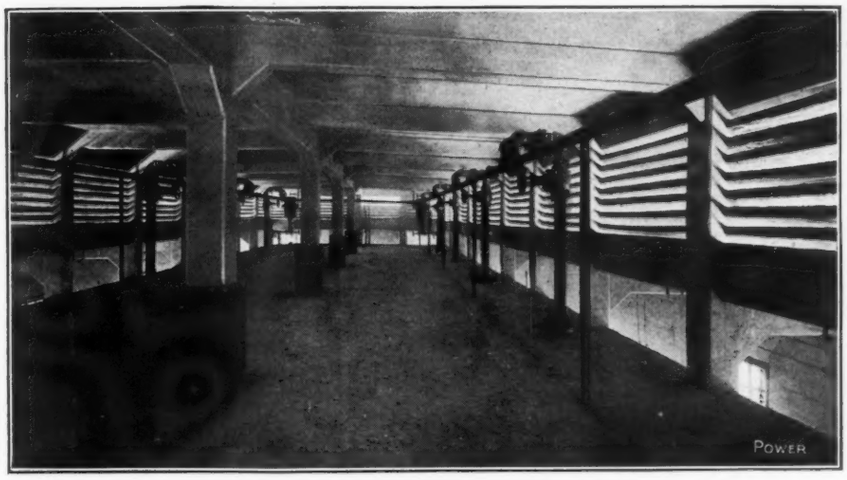


FIG. 3. VENTILATING DECK OF PRECOOLING PLANT

The precooler chambers contain about 82,500 ft. of 2-in. direct-expansion piping, running coincident with the width of the room. The piping is set in tiers 9 ft. high, with 12 coils to each stand, as shown in Fig. 4.

Two Buffalo Forge fans, with wheels 85 in. in diameter by 27 in. wide, are installed in each precooler house and are used as exhausters. The capacity of each pair is approximately 72,000 cu.ft. of air per minute, under a pressure of 3 oz. and operating at a speed of 380 r.p.m. The fans are placed in wood housing, as shown in Fig. 5, and are driven by Westinghouse motors. These fans are connected to a main 78-in. overhead exhaust duct by 58-in. No. 18 galvanized iron verticals, as indicated in Fig. 6; these pipes are similarly covered with a wood housing, Fig. 5.

The main 78-in. exhaust pipe is made of No. 18 galvanized iron; at the icing dock, where it branches into two longitudinal pipes parallel with the tracks on either side of the building, the size is reduced to 64 in. and uniformly tapers to 24 in. at the end connections. This will be noted in the right half of the plan, Fig. 2, which indicates the exhaust line, and in the cross-section, Fig. 6. The down pipes to the two bunker openings in the refrigerator car are 20 in. in diameter and of similar material, heavily insulated. For a distance of 8 ft. the ends are made of flexible canvas, strongly reinforced with iron rings, to afford adjustment to the varying position of the car. To fit the bunker openings, a wood frame with canvas and tubing on the outer edges is attached to the end of the canvas pipe; when placed in position, this is inflated by means of compressed air to render a tight connection.

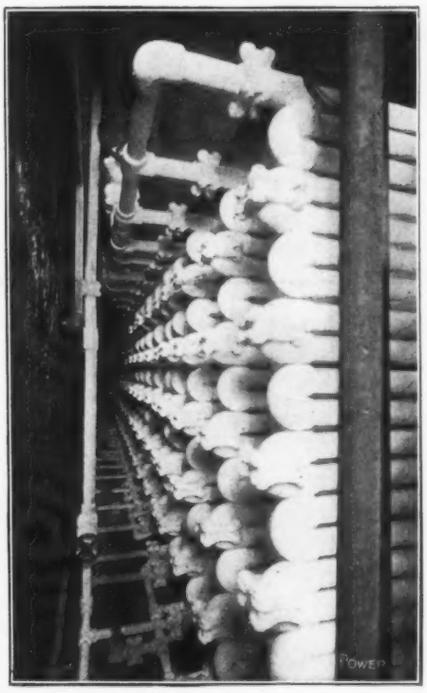


FIG. 4. DIRECT-EXPANSION PIPING IN PRECOOLER CHAMBER

The inlet or cold-blast pipe, of No. 18 galvanized iron, leading from the lower coil chamber is 68 in. in diameter. At the icing dock it is reduced to 51 in. and tapers to 28 in. at the end of the main. This is shown at the left half of the plan, Fig. 2, which indicates the inlet lines. At each respective opening, the line is fitted with a large rectangular terminal of a size to suit the side-door opening of the refrigerator car. This ter-

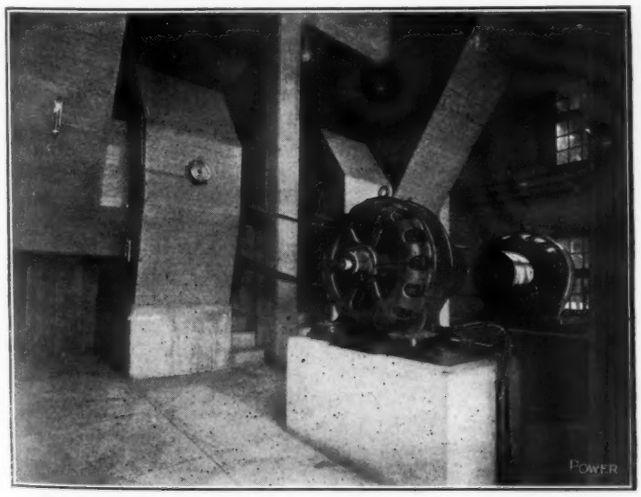


FIG. 5. MOTOR-DRIVEN EXHAUSTERS IN WOOD CASING

FIG. 7. BUTTERFLY VALVES CONTROLLING FLOW OF AIR

minal, or blast door, is of wood framing with an opening about 12 in. long at the top of the interior side, with respect to the car, the opening having curved sheets of galvanized iron to act as deflectors for the strong blast of air which would otherwise enter the car and chill its contents. The edges of the door are made of heavy canvas and interior tubing, to be inflated and allow of an air-tight connection with the car, as in the case of the bunker openings.

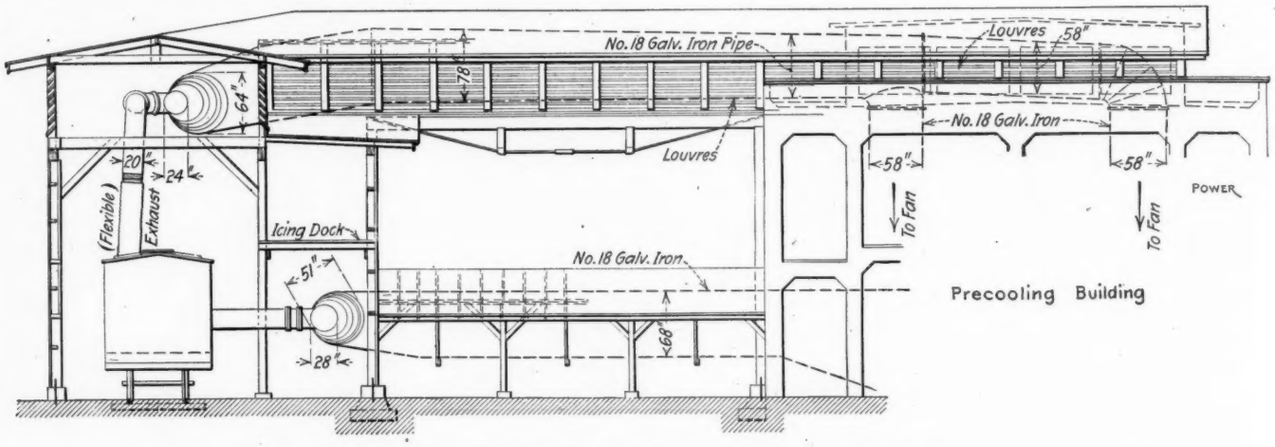


FIG. 6. VERTICAL SECTION THROUGH PRECOOLING BUILDING

OPERATION

At the end of the precooler coil chambers cast-iron butterfly valves, 48 in. in diameter, are placed as shown in Fig. 7; they operate automatically and control the flow of air through the ducts and cars. The complete details are shown in Fig. 8.

The outlet duct is fitted with automatic valves at each respective opening as indicated in Fig. 2, which work in unison with the butterfly sections in the precooler house. This automatic operation is known as the intermittent process and is the invention of A. R. Sprague; it is from this feature that the system takes its name.

The intermittent device closes the inlet to the car and allows the fans to exhaust all the warm air and gases coming through the butterfly valves from the commodities being precooled. As soon as the butterfly valves close, the inlet valve opens and forces the cool, dry air through the ducts and into the cars. This operation is repeated automatically about once every minute during the first hour of precooling. During this time about 2½ in. pressure shown by water gage to approximately 2½ in. vacuum is obtained. Through the remaining period of the process the exhaust enters the precooler coil chambers, instead of being discharged into the open. Here it is circulated over the coils, deposits its moisture and is cooled, thus allowing the later cold blasts to the cars to be thoroughly dry. Through this time the vacuum is produced at intervals of from 5 to 15 min., depending upon the temperature conditions within the car.

The cold blast ranges from 2 to 3 deg. below zero to about 15 deg. above. The walls of the car are cooled to a temperature of approximately 25 deg., permitting longer preservation of the initial icing and allowing the car to travel a considerable distance before re-icing is necessary. Four hours are required to reduce the temperature of a carload of citrus fruit from an average of about 78 deg. to an average of about 40 deg., while a car of deciduous fruit or vegetables demands about 3 hr. This time could be reduced if fruit shippers would space their loads so as to permit of more free circulation of air. All temperatures are taken in the center of the fruit and the center of the packages.

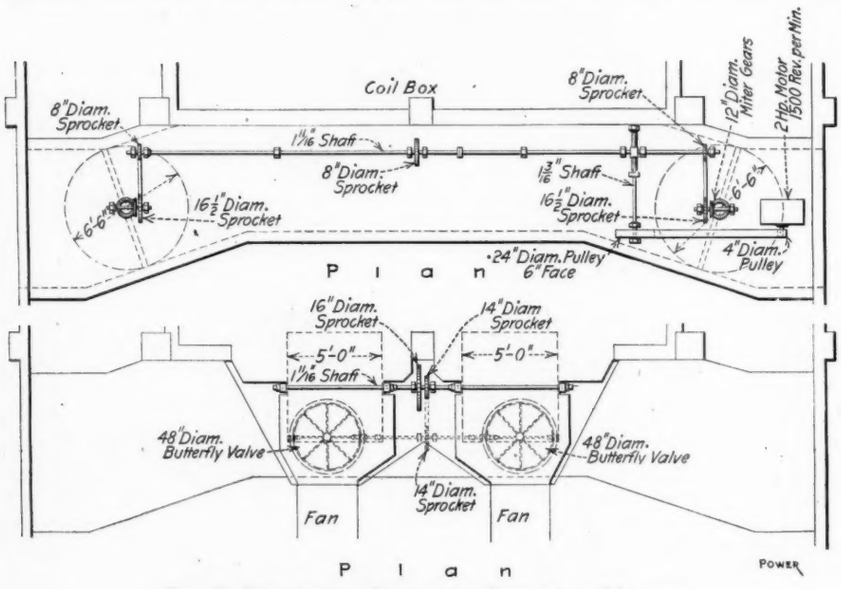


FIG. 8. MECHANISM OPERATING BUTTERFLY VALVES

Many experiments have been made to determine the exact conditions with and without the operation of the intermittent vacuum device. When it is not used and a low temperature is retained on the inlet to the car, the commodities being precooled are liable to freeze. The device permits of a much more equal distribution of air throughout the car than could possibly otherwise be obtained, even though baffles were used.

An icing dock is constructed in the train shed at a height closely approximating the level of the top of a refrigerator car and allows of ready access for filling the car bunkers with ice after the precooling has been completed. The platform is equipped with an endless-chain conveyor for handling the ice and six unloading points thereto are provided leading from the storage chambers in the main building.

The company has expended about $800,000 in developing the Colton plant, and the system has proven so successful that it has erected a similar plant, of slightly less capacity, at Roseville, Calif., to handle the deciduous products from this section of the state.

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POWER

Turbine or Engine

What is an engine and what is a turbine?

There are those who will respond that all turbines are engines, but all engines are not turbines.

All right, then. When does an engine cease to be a turbine? What is the difference between a turbine and a rotary engine; not in the general but in the particular sense in which the word engine is used?

In the engine, used in this sense, the steam or other working fluid is admitted to a confined space. The molecules of which it is composed fly about in this space and by their bombardment of the containing surfaces produce pressure. Some of the surface, as the piston of a reciprocating engine, moves under the pressure and the work produced is the product of the change in volume and the average pressure during the change.

If steam is permitted to flow without obstruction from a region of higher to one of lower pressure, the energy developed will be applied to getting up velocity in the moving steam and will be present in the stream or jet as kinetic energy. To get the energy out of it the steam must be brought to rest.

A turbine, as we understand it, is a device for doing this. Either by reaction upon the nozzle in which it expands or by impact upon and change of direction by contact with moving surfaces, the fluid mass is brought more or less completely to rest and its energy of motion is appropriated by the device constructed for that purpose; that is, the turbine.

In the engine the pressure of the operating fluid acts directly upon the moving surface, and expansion is controlled by the yielding of that surface under pressure. Kinetic energy comes into consideration only when the analysis goes into the constitution of the steam. The only velocity involved is molecular; the fluid mass simply undergoes change of volume. It is analogous to a weight driving a clock, a restrained, controlled lowering of the weight with a direct conversion of its potential energy into work, or to a hydraulic press pushing a hub on a shaft with a pressure directly applied and moving only as fast and as far as the load moves.

In the turbine the steam acts not by molecular impact but by mass impact; not by static pressure but by momentum and inertia. The pressure acts first upon the working fluid itself and gets up velocity, momentum in it, and then that momentum is absorbed by the turbine. It is analogous to a pile driver, the weight of which gathers momentum in its fall and is brought to rest by impact with the pile upon which the work is to be done.

He who writes definitions hunts trouble, and we do not presume to draw lines where perhaps boundaries should be shaded, but the above seems to us to state the real distinction between the pressure engine, rotary or otherwise, and the turbine.

There has been some discussion in engineering circles as to which class belongs the Spiro motor, described in our issue of Feb. 6. Perhaps it is in the shady borderland between the two. We should be glad to have expressions of opinion from our correspondents.

The San Antonio Report

Some two months ago we published an account of the San Antonio locomotive boiler explosion which resulted in the death of twenty-six and the injury of thirty-two people. In this issue appears an abstract of a noteworthy report on this terrible disaster made by the inspector of locomotive boilers and two assistants to the Interstate Commerce Commission. The report is unusually complete and thorough and is unique among boiler-accident "post-mortems" in that it does not saddle the blame on the killed but puts it where it rightly belongs.

As the report states, the sling stays attaching the crown bars to the wrapper sheet were of wrought iron where the specifications called for steel, and were

attached with one-inch bolts where one-quarter inch larger should have been used. The crown bars, not being supported on the sides of the firebox as in the older type of crown-bar boilers, all the strain was on the sling stays, making it all the more important that the larger bolt should have been used.

Subsequent tests by the United States Bureau of Standards showed that the stays with the one-inch bolts had a factor of safety as low as 2.67, considerably below the recognized standard. With the larger bolt the factor of safety would have ranged from 3.67 to 4.15.

The evidence developed that turning down the adjusting screws of the safety valves and a defective gage permitted pressure exceeding that the boiler could endure with stays of such low factor of safety. It was also shown that the rules governing locomotive-boiler inspecting were disregarded by the company's employees assigned to make and certify to the reports required by law.

A report stating that the safety valves had been set was sworn to on Mar. 16, whereas this work was being done when the explosion occurred, and in addition it was proved that the inspector had failed to witness the inspection of the steam gage, although he certified under oath that this work had been done.

The railroad company was severely censured for permitting such reports and inspections to be made, and for allowing such important work as setting a safety valve to be conducted by an employee of whose experience or judgment, the testimony showed, it was utterly ignorant. The company was also blamed for keeping a boiler in service whose factor of safety was below standard.

In the opinion of the inspectors, the laws governing the inspection of locomotives and the setting of safety valves were sufficiently comprehensive for safety if intelligently complied with, but as an extra precaution action was taken to compel using a second gage always in view of the person setting the safety valve, and cleaning the siphon pipe and its connections every time the gage is tested.

With these additional inspection requirements, locomotive boilers should be well guarded and the danger from explosions reduced to a minimum, but of what advantage are the most stringent laws if they are not properly complied with or enforced by the proper officials? Under similar conditions other accidents, equally as bad or even worse, might readily occur. If safety is desired the laws must be rigidly enforced. The inspector must be competent and, above all, trustworthy. Ignorance of his qualifications does not relieve his employer of blame, and a company guilty of criminal negligence should pay the full penalty of the law.

What kind of inspection will allow a boiler to be operated with a factor of safety so much below the standard and, worse still, allow a man to tamper with a safety valve who evidently was so ignorant of the requirements or precautions to be observed? We have not heard of any penalty inflicted for such a flagrant case of negligence and disregard of the spirit of the law, but if the death of twenty-six human beings is not enough to incite some action stronger than censure by an inspector, then all the laws that have been and could be formulated on boiler inspection will be of very little use.

The laws must be enforced and the punishment made compatible with the infringement, or just such disasters as that at San Antonio will continue to discredit engineering practice in this country.

Fallacy of Submerged Flotation

The recent loss of the "Titanic" has revived the old discussion as to whether objects sinking in great depths of the ocean go clear to the bottom. That there ever could have been any belief that they do not, seems attributable only to our greater familiarity with the action of bodies floating in the air, and our failing to recognize the incomparability of air with water flotation.

They are incomparable for the reason that air does exhibit variation of density with height due to its compressibility, while to all intents and purposes water is incompressible; hence its density does not sensibly increase except at tremendous depths, and then not sufficient to support objects which have sunk entirely beneath the surface.

Even considering a sinking body of closely the same specific gravity as water, it is inconceivable that its structure could resist the enormous hydrostatic pressure at great sea depths sufficient to prevent its specific gravity from increasing faster than that of water as it continues to sink. Therefore, if anything, a body sinks faster the deeper it becomes submerged.

In this connection the letter under the same title on page 816 is of interest as it is from an authority on hydrostatics and gives a most lucid explanation why the submerged flotation theory is untenable.

Grate Distance from Boiler

How far should the grate of a return-tubular boiler be placed from the boiler shell?

In common practice, the distance seems to vary from 24 in. for anthracite coal to 40 in. for bituminous, with 36 in. as a good average, according to the coal used. This distance is assumed to allow ample space for the thorough mixing of the gases and for producing complete combustion.

What happens when a furnace is improperly designed, in that the grate is too close to the shell, may be shown by a very simple experiment. Place a tin can over a gas jet and too far down into the flame and a similar condition exists. The metal will become coated with soot, showing that incomplete combustion has taken place, caused by the cooling of the gas.

When the boiler is set too close to the grates, the flame impinging on the fire sheet is cooled before the combustion is complete and dense black smoke is formed which issues from the stack. Generally this smoke is attributed to lack of air, and probably many times that is the true cause, but the boiler shell too close to the grates would unquestionably have the same effect.

The results obtained from dutch-oven furnaces show that the gases of combustion should be thoroughly ignited before passing to the combustion chamber. All in all, this qestion is one worthy of the attention of those concerned: "Should the grate be farther from the boiler shell?"

In the accomplishment of many things there are several ways to perform them, but the boiler room with only one exit offers only one means of escape. Does this include your plant?

Test of Cylinder Oil

By A. E. Walden

A given oil that will work well in one plant will not do so in another, even if the engine size, speed and pressure are the same, due to many conditions such as the structure of the iron entering into the cylinder, the method of the application of the oil, the acid or alkaline condition of the steam, etc.

In one plant under the writer's charge oil was first purchased costing 70c. per gallon; this was changed later to 50c. per gallon with the same operating results, and in neither case did the intermediate cylinder indicate oil by the use of filter papers or otherwise, yet it was always smooth and glossy. After trying several oils one was finally secured that cost about 30c. per gallon and actually reduced the oil consumption to one barrel per day. Specific gravity, flash and fire test are not always to be relied upon as an indication of good oil.

The test records, Figs. 1 and 2, show these determinations for two oils that were selected after numerous operating tests had been made and of which a sample, No. 75, was selected for further test. This oil from the specific-gravity test answered the purpose for a short time only, a certain time apparently being required to cut out the old oil and remove the polish from the cylinder. This cylinder had been lubricated with one-half the amount of oil, known later as No. 100, to keep it from knocking that was required with the No. 75.

With No. 75, Fig. 1, the flash point was so low in the friction-testing apparatus that it took fire and the test could not be carried out as was indicated for No. 100. Both oils cost 30c. per gallon.

temperature, as was shown after superheaters had been installed and the temperature raised for some time from 385 to 425 deg., the operating pressure being about 125 lb.

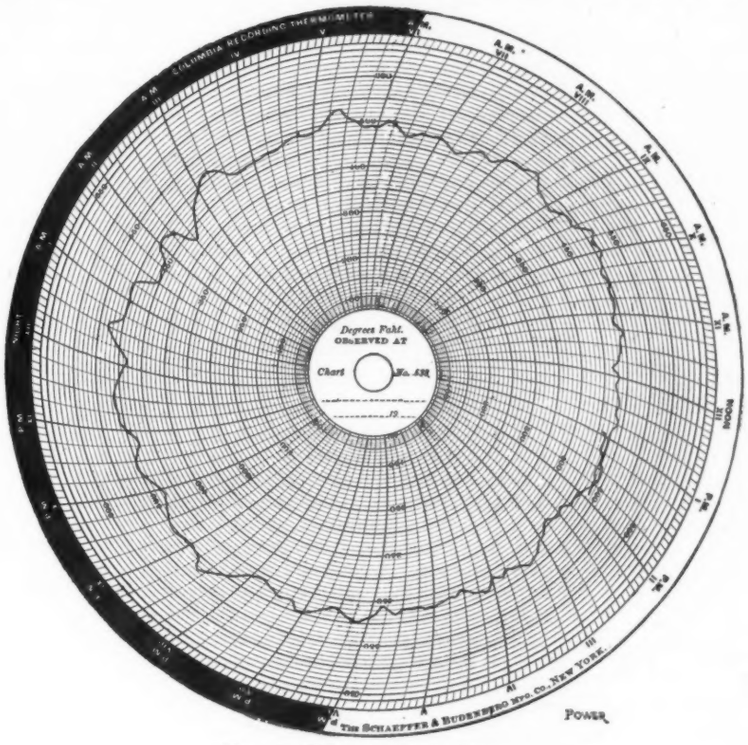


Fig. 3. Temperature Chart

No trouble has been experienced, and the quantity of oil has not been changed, which is eight drops per minute for No. 100. It required from 24 to 26 drops per minute of No. 75 to keep the valves quiet on this unit, which was a Corliss pumping engine having a 16x30-in. high-pressure, and two 24x30-in. low-pressure

so as to secure a uniform rise, as some oils, if the temperature is raised rapidly, will have the flash point reduced. The temperature rise should be made at about 20 deg. per minute, applying the test flame every 7 to 10 deg., but experience will determine the exact rate of rise.

Among the friction-testing machines there is one with bearings having a loading capacity of 6000 lb. and provided with a thermometer and pipe connections for supplying steam. This machine maintains the temperature in the bearings at

#75
OIL ANALYSIS.

MAKE OF OIL #75 DATE
KIND OF OIL Cylinder COLOR Red & Green
DENSITY AMOUNT OF FEED OF OIL C. C.
VISCOSITY AT 70°F. 160 212°F. ASH %
FLASH POINT 560 °F. FIRE TEST 605 °F.
COEFFICIENT OF FRICTION AT °F. TARR None %
COLD TEST 45 °F. SOAPY MATTER %
LOST BY EVAPORATION 12 HOURS AT 140°F. % SP. GRAVITY 25.4
ANIMAL ACIDITY None TALLOW 6.00 %
MINERAL ACIDITY None □ FT. RUBBING SURFACE
NO. REVOLUTIONS PRESSURE IN RUBBING SURFACE LBS. PER □ IN. 50/100
NO. FT. TRAVELLED BY RUBBING SURFACE
ELEVATION OF TEMP. °F. ABOVE INITIAL TEMP. OF °F.
CHEMIST

Fig. 1

#100
OIL ANALYSIS.

MAKE OF OIL #100 DATE
KIND OF OIL Cylinder COLOR Red & Green
DENSITY AMOUNT OF FEED OF OIL C. C.
VISCOSITY AT 70°F. 182 212°F. ASH %
FLASH POINT 560 °F. FIRE TEST 600 °F.
COEFFICIENT OF FRICTION .054.06 AT 350 °F. TARR None %
COLD TEST 47 °F. SOAPY MATTER %
LOST BY EVAPORATION 12 HOURS AT 140°F. % SP. GRAVITY 26.0
ANIMAL ACIDITY None TALLOW 6.20 %
MINERAL ACIDITY None □ FT. RUBBING SURFACE
NO. REVOLUTIONS PRESSURE IN RUBBING SURFACE LBS. PER □ IN. P/100
NO. FT. TRAVELLED BY RUBBING SURFACE
ELEVATION OF TEMP. 350° °F. ABOVE INITIAL TEMP. OF °F.
CHEMIST

Fig. 2

Results of Oil Analysis

As shown by Fig. 2, the temperature specified in the test was to be 410 deg. This was not reached because of the type of thermometer used, but a temperature of 350 deg. was attained and the oil was and is able to stand a much higher cylinders. When the steam temperature was raised by means of superheaters, the amount of oil was not changed and the cylinders are in good condition.

In making the flash and fire test, the oil temperature was raised very slowly, the same point as the oil would be used in the cylinders, and at as near the test conditions as possible. From the writer's experience, however, there appears to be a field for a good cheap steam-heated friction-testing machine.

Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Center Crankpin Oiling Device Improved

The crankpin on a new high-speed center-crank engine would heat without apparent cause. It would run nicely for a week or more and the engineer would feel that no further trouble was to be expected, when suddenly it would heat and smoke until it required very prompt action to prevent the babbitt metal from melting out. As the engine is of the inclosed type, the cause of this trouble was not promptly discovered. Examination showed that the brass ring into which the oil for the crankpin is delivered was not deep enough to hold sufficient oil for the required service.

As may be seen in Fig. 1, oil from an elevated tank is delivered through sight-feed *A* and nozzle *B* into the swiftly revolving ring *C*, but only a small part of it goes to the crankpin, while the remainder is thrown out of the ring against the oil guard. Undoubtedly the best remedy for this defect was to remove the ring *C* and substitute a new one, but this was out of the question on account of the labor involved.

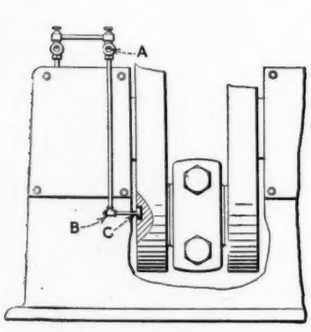


FIG. 1. CRANKPIN OILING DEVICE

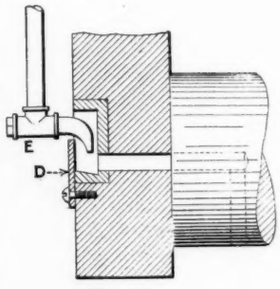


FIG. 2.

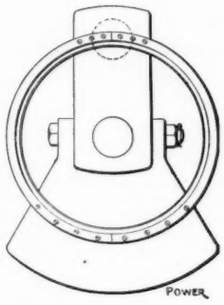


FIG. 3.

It was decided to add a ring cut out of sheet brass, the effect of which is shown at *D*, Fig. 2, thus making the cavity deep enough to fully cover the hole through which oil must flow to the crankpin. It was made in halves and secured in place by 14 screws, as shown in Fig. 3. For about one-half of the circumference it was impossible to use screws, because there was not sufficient metal in the original ring to hold them; hence the two metals were soldered together, and an oil-tight joint secured. It is practically impossible to put on such a ring so that it will run true; therefore it was cut smaller on the inside than required.

A cross feed from a lathe with a tool post carrying a suitable tool was fastened to one of the main bearings, the engine run at its lowest speed, and very light cuts taken from the flat brass ring until it was so enlarged as to admit the oil nozzle without friction, as shown in Fig. 2; this secured a perfectly true circle.

The oil pipe shown in Fig. 2 is fitted with a tee at *E* instead of an ell. By removing the plug shown, a wire can be inserted to clean out any foreign matter in the tee and nozzle.

W. H. WAKEMAN.

New Haven, Conn.

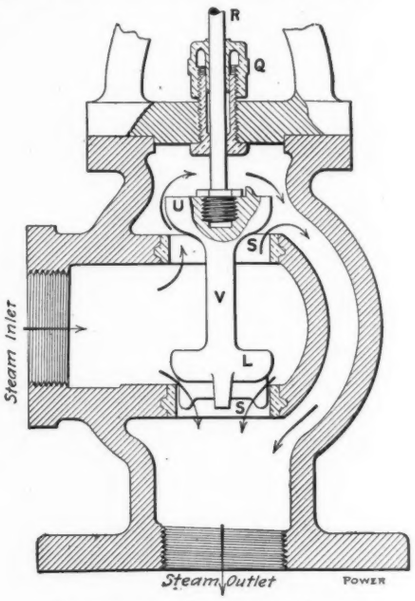
Remodeling a Defective Governor Valve

The accompanying illustration represents the valve-chamber and the valve of an automatic throttling speed governor. In experiments made, I found that the outside valve actuating mechanism (which automatically moves or holds the valve *V* in position in relation to the valve seats *SS*, thereby regulating the quantity of steam allowed to enter the steam chest of the engine) was considerably disturbed by the fluctuating steam-chest pressures acting on the valve stem *R* at the stuffing-box *Q*, especially at high steam pressures.

Circumstances compelled me to seek a means of neutralizing the effect of the constantly varying force exerted at this point, which caused the governor to dance, and sometimes to race. The valve in action is supposed to be automatically held in position in relation to its seats by maintaining the balance of two opposing forces, viz.: the centrifugal force of revolving weights and the centripetal force of springs under tension. Disturbing the balance of these forces will cause the valve to be moved, according to the direction of movement of the greater force.

The steam-chest pressure exerted against the valve stem at the stuffing-box *Q* tends to disturb the action of the above forces.

If this third force be of constant amount it will have no ill effect on the regulation, but as the steam-chest pressures of all throttle-governed engines vary considerably, it causes trouble with the regulation.



THROTTLING GOVERNOR VALUE

The following explanation will show how this force was made constant for a given boiler pressure: The top piston *U* of the valve was made larger in diameter than the bottom piston *L*, the increase in cross-sectional area amounting to the exact equivalent of the valve stem *R* cross-section. Thus when the valve *V* closes the governor ports there will be an upward force exerted upon the valve-actuating mechanism (not shown) of 13 lb. (boiler pressure 175 lb.) from the steam pressure acting on the differential surface of the top piston *U*. When the valve is fully opened and the boiler pressure is on the delivery side of the valve, there will still be 13 lb., but in this case from the pressure at the stuffing-box.

With 175-lb. boiler pressure, and with varying pressures on the delivery side of the valve, the sum of forces from the top piston and the valve-stem stuffing-box always equals 13 lb., increasing at one of these points in proportion as it

decreases at the other. This amount varies only as the boiler pressure rises or falls. The tendency of the governor to dance or race from the effect of fluctuating steam-chest pressures has thus effectively been overcome.

L. J. WATSON.

Port Huron, Mich.

What Should Be the Crank Positions?

At a large pumping plant where I am employed, we have two cross-compound horizontal Corliss pumping engines with unusual water ends in that they are designed as if driven by a cross-triple steam end. The intermediate crosshead has no steam cylinder, being driven by a connecting-rod from the intermediate crank on the shaft, which has two flywheels. What relative position should the cranks have to each other?

If there were three steam cylinders, I would say the cranks should be 120 deg. apart, but as the intermediate crank has no steam cylinder it receives energy from the shaft and flywheels; therefore I believe that the high- and low-pressure cranks should be set at 90 deg. and the intermediate at 135 deg. Perhaps other readers will express their opinions on this subject.

C. E. ALDRICH.

Tranquility, Calif.

Continuous Flue-Gas Sampler

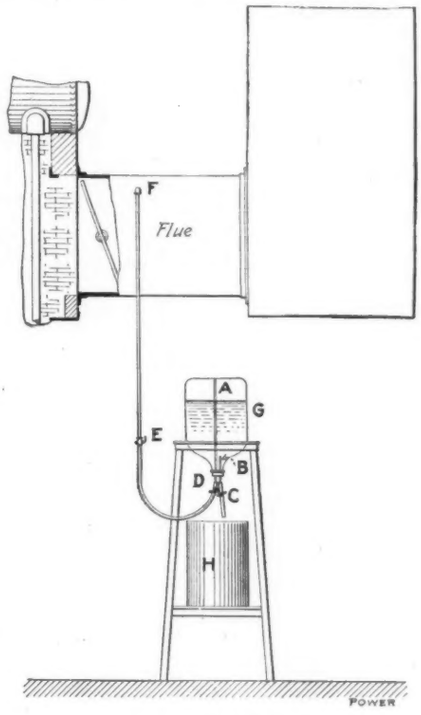
The accompanying illustration shows a simple but effective device used to obtain an average flue-gas sample for each day's run. The outfit may be constructed at small cost and it works very satisfactorily.

It consists of a 5-gal. glass bottle, *G*, mounted upside down on a wooden framework. Underneath the bottle is placed a metal can of about 6 gal. capacity. At a point in the flue, as at *F*, is inserted a ¼-in. iron or brass pipe reaching half way across the flue, with its open end so placed as to be directly in the path of the outgoing gases but not to interfere with the action of the damper. A preferable aspirating tube would be one extending all the way across the flue, and perforated with ⅛-in. holes every 6 in. apart, the holes being placed down so as to prevent the tube filling with soot.

A ½-in. pipe leads from *F* down to the level of the bottle with an air cock, *E*, at the bottom. Connection to the bottle is made by a heavy rubber tube, which may be closed by a pinch-cock, *D*. Into the mouth of the bottle is fitted a rubber cork having two holes, into which are inserted a ½-in. brass pipe *A* reaching to within $\frac{1}{16}$ in. of the bottom of the bottle, and into the other a short piece of pipe shown at *B*. On the outer end of pipe *B* is a short rubber tube which is fitted with a pinch-cock, *C*.

To use the sampler, fill the bottle with water, attach the short rubber tube, close the pinch-cock *C* and invert the bottle. The long rubber tube *D* is then connected to the cock *E* and the pinch-cock *D* opened. The pinch-cock *C* is then opened slightly so that the water in the bottle flows out one drop at a time into the pail *H*.

The gas will be slowly drawn from the flue into the bottle, and by regulating *C* so that the bottle empties in 8 hr. an average sample for the day's run may be obtained for analysis in an Orsat



FLUE GAS SAMPLER

apparatus. Before taking a sample of the gas, the air contained in the pipe and tube *E* should be withdrawn by producing a vacuum in it. The success of the sampler depends upon having all the joints air-tight.

B. S. NELSON.

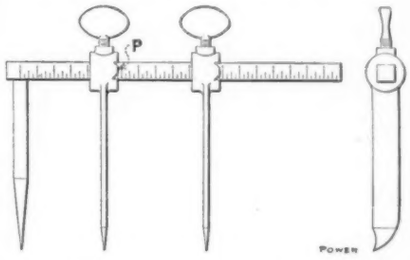
Bayonne, N. J.

Home-Made Gasket Cutter

The accompanying illustration shows a home-made gasket cutter which has given much satisfaction in our plant.

The beam was made of ½-in. square steel with a pointed leg secured at one end. The knives were made of tool steel with a square hole and thumbscrew in the hub as shown. After the knives have been fitted to the beam and ground, secure one to the beam and scribe a circle; then place the other at exactly the same place on the beam, with regard to the pointer *P* and bend the knife so that it will scribe a circle of the same diameter as the first. This is necessary in order to have both knives cut true to the same graduations.

As each inch on the beam means two in the diameter of the gasket, the half-



GRADUATED GASKET CUTTER

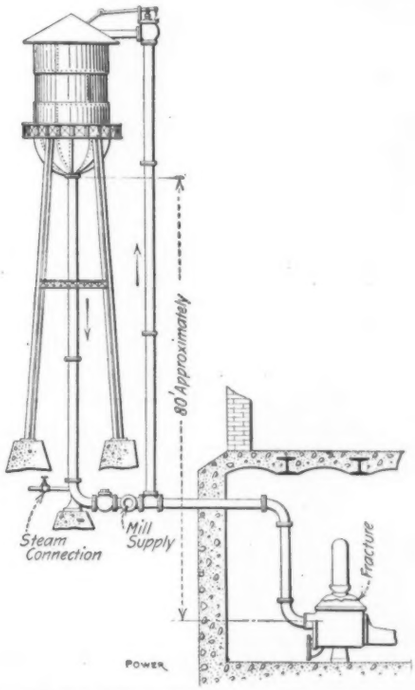
inch lines should extend all the way across the beam and read 1, 2, 3, etc.

O. O. LEMON.

Bristol, Va.

Impact of Water Broke Pump

A rather peculiar break happened in our plant recently. The pipe from the supply tank was not sufficiently protected where it is connected to the tank, and in the cold weather it would freeze at



PIPING ARRANGEMENT OF FRACTURED PUMP

this point. As this was a rather difficult place to get at, I decided to put steam in at the bottom of the pipe in order to thaw it. It worked successfully and we had occasion to make use of it some 8 or 10 times during the extreme cold weather.

Referring to the illustration, it will be seen that the pipes to and from the tank are on one line, but the check valve prevents the water from entering the tank at

the bottom. When the steam is turned on, the pump is run slowly, the pressure keeping the check valve closed, so that the steam has no outlet except through the tank. During the last cold spell the pipe was again frozen; the steam was turned on as usual, but in a short time there was a violent crack and the basement was deluged with water. It took but a minute's work to locate the trouble and get the water shut off; then I found that the air chamber, which is cast integral with the valve bonnet, had been broken off, and also the greater portion of the bonnet. As to what caused the break, we must conclude that the pump had stopped, or at least was going so slowly that the water receded, leaving the pipe empty.

The ice gave way suddenly, allowing the water to drop in a body, producing a ramming effect when brought to a stop at the pump. It was evident that considerable pressure was exerted, as the casting was fairly good.

S. M. QUINN.

Hudson, N. Y.

Saving Pump Rods

There are conditions under which it is impossible to prevent piston rods from scoring, and this is especially true with a steam pump of the type and size usually found in the average power plant. At best, the steam pump receives but little attention on account of its reliability and extreme simplicity, and often the gritty nature of the liquid pumped will flute a piston rod in a short time.

I once had a duplex receiver pump which was used to take the return water from a number of machines and elevate it to a tank a short distance above it where the water was used to supply the boilers and for other purposes.

As all of this equipment was new there was considerable sand, scale and other grit in the pipes and machines and in time this came back to the receiver and then to the pump, where it soon scored the steel rods at the water end. As the pump was in use almost constantly, all we could do was to pack the rods as often as was necessary and wait for the time to make the repairs.

The steam ends of the rods being in good condition, I thought that bronze rods could be substituted at the water end, but the construction prevented this. I found some 1⅛-in. brass tubing which was of the same outside diameter as the piston rods. After the rods were removed from the pump they were put in the lathe and turned down on the water end to 1 1/16 in., beginning at the taper and continuing to about a half inch from the crosshead. I then lubricated the rod with white lead and started the tubing, and by ramming it gently against a post I soon had it up about an inch from the shoulder, when I put it in the lathe and cut the end of the tubing off where it was to fit against the shoulder at the crosshead end.

Then after driving the tubing home, the other end was cut off in like manner so that when the water-piston head was snug up on the taper it would also be snug against the end of the tubing.

To facilitate repairs, both rods were turned so that both pieces of tubing would be the same length, and a record of this was entered in the log for future reference. In case of renewal it is only necessary to split the old tubing to remove it, slip a new piece of the required length on in place of it and the rod is as good as a new bronze rod.

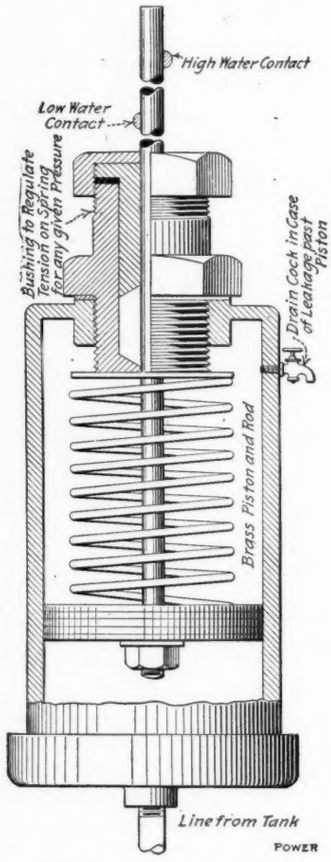
The cost of the tubing for this job was 35c., and the pump was out of service only about six hours.

G. H. WALLACE.

Racine, Wis.

High and Low Water Alarm

The accompanying illustration shows a device for indicating the quantity of water in our sprinkler tank. It is used both as a high and low water alarm,


High Water Contact
Low Water Contact
Bushing to Regulate Tension on Spring for any given Pressure
Drain Cock in Case of Leakage past Piston
Brass Piston and Rod
Line from Tank
POWER

SHOWING MECHANISM OF ALARM

and is not so liable to disorder as the common wire-and-float arrangement. The device should be mounted in a conspicuous place in the engine room so as to be readily seen by the engineer. The weather does not affect it as is the case with arrangements exposed to low temperatures where ice gathers on the pulleys and chains, preventing satisfactory operation.

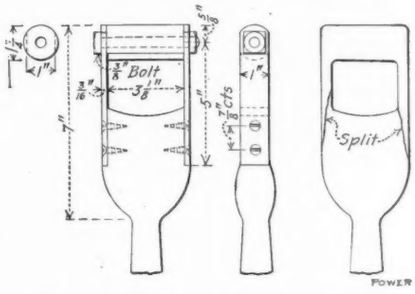
The device may be set at any desired pressure and made from pipe fittings. Its cost is very low and the advantages from its use more than compensate for the cost of the installation.

H. PREW.

Montreal, Can.

Repairing a Scoop Handle

The accompanying illustration shows how a scoop handle was repaired that had broken as shown in Fig. 2. The side pieces are of mild steel, but wrought iron will also answer the purpose. They are held in place by screws, as shown, but before the screws can be driven in the hardwood handle, it is necessary to



SHOWING REPAIRS TO SCOOP HANDLE

drill holes a little smaller in diameter than the screws.

The grip or roll is made of maple turned up in the lathe, with a hole through the center for a ⅜-in. bolt. Two opposite sides of this roll are planed almost flat, as shown by the dotted lines in the end view of Fig. 1. It is then finished with sandpaper.

FREDERICK L. JOHNSON.

Paterson, N. J.

Some Good Kinks

When renewing small round disks from globe valves, do not try to dig out the old ones. Just hold the disk in or over a flame—a match, or gas get, or torch—until it is softened by the heat; then it may be dug out with ease.

When packing valve-stem or other stuffing-boxes with twist packing, do not use a knife to cut it off, for it is next to impossible to do this without disarranging the packing and fraying the ends; use a pair of stout, sharp shears or a pair of snips.

When filing for a finished surface it is exasperating to have a bit of metal imbed itself in the file and on the next stroke dig a deep groove. To prevent this, clean the file carefully and give the entire face a coating of chalk. If cleaned and chalked from time to time, it will be found easy to get a finely smoothed surface.

When forced draft is used with some kinds of water-tube boilers, ashes blow out between the headers and occasionally sift down through the crevices around the door. This has been obviated in some cases by filling in between the headers with asbestos.

One morning an engineer discovered that the lower bolt on the wristpin wedge of the Corliss engine was broken. The wristpin had to be backed out, the crosshead moved away, the old bolt removed and a new one put in, and then the whole thing assembled and adjusted. By dint of hustling and some intelligent assistance he was ready to start in 10 minutes after the discovery. Strange to say, no small part of the aforesaid intelligent assistance was rendered by a person at that time many miles away. A former engineer had taken pains to slot the ends of all wedge bolts, so that in such an emergency they could easily be backed out.

When setting up on a gland, it is customary to adjust it so that it shows equal opening all around the stem or rod. This perhaps is satisfactory, if it is known that the rod is central. However, in the case of a rod that is not central—for instance, a rod of an engine whose piston is down—if the gland is adjusted to show equal opening on the outside, there is danger of making the gland rub on the inside. The safest way in all cases is to use a feeler to see that the gland clears the rod at all points. A feeler made of ordinary pipe covering band has been used with success.

WILLIAM E. DIXON.

Malden, Mass.

Air Pump Diagrams

The accompanying indicator diagrams, Figs. 1, 2 and 3, were taken from an air pump used in connection with a vacuum-cleaning system. The cylinder has a 19-in. bore and a 12-in. stroke and is driven 150 r.p.m. by a 25-hp. motor. A 40-spring was the only one available for taking the diagrams; and it was possible to connect to one end only of the cylinder. The inlet valves are of the Corliss type driven by a crank outside the flywheel on the end of the main shaft; there are five discharge valves of the poppet type.

The diagram shown in Fig. 1 was taken with the machine running at full speed and the cleaning-out valve on the dry separator open. The suction pressure being atmospheric, less pipe friction was present, and the only work done was that required to compress the air enough to open the discharge valves as shown at *X* in the diagram. The drop in pressure at the beginning of the suction stroke *Y* is caused by the suction valve not opening exactly on the center.

The diagram, Fig. 2, was taken with the machine carrying its regular vacuum —no cleaning tools were used—the automatic unloading valve being open just enough to prevent the vacuum from going above the proper point. When in good condition this valve should regulate within 2 in., but here it showed a variation of about 8 in. and a vacuum of 20 in. on the mercury column. There appears to be very little work done at this time as the compression line follows the suction line back over one-half the stroke.

The diagram, Fig. 3, was taken while

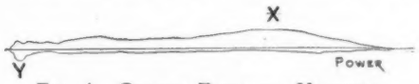


FIG. 1. CRANK END; NO VACUUM

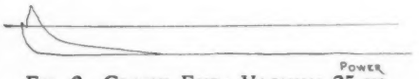


FIG. 2. CRANK END; VACUUM 25 IN.

FIG. 3. CRANK END; VACUUM 12 IN.

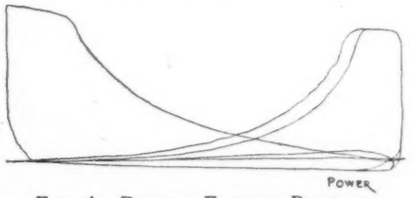


FIG. 4. BROWN ENGINE DIAGRAM

using the cleaning tools, the working vacuum being about 12 in., as shown by the mercury column, and indicating that a large volume of air was being pumped.

The diagram shown in Fig. 4 was taken from a 16x42-in., four-valve, flat, multi-ported Brown engine, each set of valves being driven by a separate eccentric; the camshaft has been in service over 30 years. The engine was running badly at the time of taking the diagram. Will some interested reader tell me what should have been done to properly set the valves?

J. CASE.

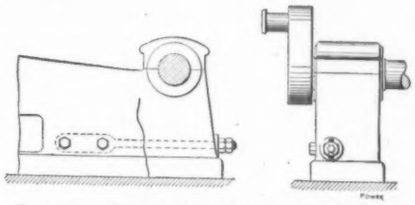
Hyattsville, Md.

Engine Bedplate Repair

Recently a crack became visible in the bedplate of our engine, and soon caused excessive vibration of the main bearing. We could not shut down as this was our only engine, so we made the repair shown in the illustration.

Using a 20-in. iron bar, flattened at one end and threaded at the other, we cut a slot in the front of the bedplate to allow the flat bar to pass through. As a handhole was convenient we clamped the bar to the bedplate and then drilled through the plate and the bar; the bar was then fastened with two 1-in. bolts. By tightening the nut on the end of the bar at the front of the bedplate, the broken parts were drawn together.

The entire repair was made without stopping the engine, and after three months' service in carrying the usual



CRACKED BEDPLATE, SHOWING METHOD OF REPAIR

load the crack shows no signs of further extension.

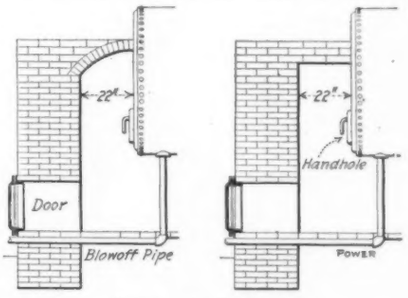
H. LOCKE.

Manchester, England.

Would the Arch Affect the Boiler's Operation?

Will some reader advise me as to the setting, form and effect of different shaped arches in the combustion chamber of a 60-in. horizontal tubular boiler?

In a plant where the original arches



FURNACE ARCHES

were circular in form they disintegrated and fell, and square arches were substituted. The illustration shows the two forms, Fig. 1 the original arch and Fig. 2 the new arch. Would the square arch have a deleterious effect on the draft and combustion in the boiler?

W. G. SEARS.

Alberta, Can.

Air Moisture Separator

At our plant there are a number of small air lines exposed to the outside atmosphere. During very cold weather the moisture in the air would freeze and give trouble; it sometimes took two days to thaw out the frozen lines.

To overcome this difficulty I made a separator out of a 5-ft. piece of 10-in. pipe, and ordinary 10-in. caps were used to close the ends of the pipe. The top cap was drilled and tapped for a safety valve and for the inlet and outlet air pipes. The inlet pipe extended about half way into the separator and the outlet was taken direct from the cap. A ½-in. line with a valve was connected to the bottom cap to drain off the condensation.

WILLIAM KELLEY.

Brooklyn, N. Y.

Questions Before the House

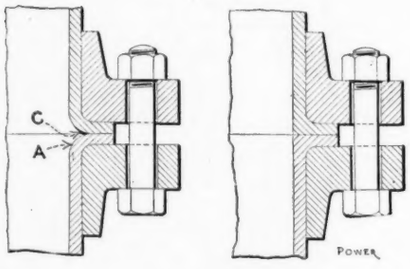
Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Reinforced Van Stone Joints

In the Feb. 27 issue, the writer of an article under the above heading takes the stand that nothing is gained in the way of efficiency in the new style, or reinforced van stone joint, basing his argument on a comparision of the theoretical tensile strength of the pipe and the combined tensile strength of the bolts.

From this point of view his argument is theoretically correct, but he has failed to consider that there are other than tension strains only to be dealt with in a van stone joint. I have never known of a flanged joint that failed due to the weakness of the bolts, while in one or two instances, at least, the old-style van stone joint has failed.

Fig. 1 illustrates the old-style joint and Fig. 2 is one type of the improved



OLD AND IMPROVED VAN STONE JOINTS

joint. There are a number of other modifications of the joint on the market, each with some particular advantage claimed for it; but the one shown in Fig. 2 overcomes all the practical objections to the old style.

In Fig. 1 the weakness of the joint is at *A*, the point of curvature. If the tension strain only on the pipe is considered, this would not be so as was proved in the article referred to, but a more careful analysis of the joint will show that all the vibration in the line is "centered" at the point *A*, with point *C* as a fulcrum. The flange as made in practice, is bored $\frac{1}{16}$ to $\frac{1}{8}$ in. larger than the outside diameter of the pipe in order to allow the flange to turn freely on the rough surface. This difference in the pipe diameter and the flange bore allows a small amount of play between the pipe and the flange, which in turn produces a vibrating strain at the point *A* caused by the constant vibrations present in all high-pressure steam lines. This constant vibration, even though it is slight, has a decided tendency to crystallize the metal at this point; the greater the vibrations the more rapidly will the metal crystallize. This defect is partially overcome by lengthening the hub of the flange, and it is simply a matter of time when the factor of safety in the original tensile strength of the pipe will entirely disappear.

Another defect in the old-style joint is the small water pocket formed over each gasket. Every practical man knows what it means to the joints to have water in the steam pipes. Where the flow of water is constant there would be no trouble as the water would be carried along with the steam, but where the flow is intermittent, as in plants operated ten or twelve hours a day, when the line is dead or the flow of steam stopped, a small amount of distilled water will collect at each joint. This will not only cause the joints to leak through the gaskets, but distilled water will attack iron and steel and cause corrosion to set in at this point.

W. LEE RAUECHE.

Birmingham, Ala.

Fallacy of Submerged Flotation

A subject which has excited the interest of many readers of the daily papers since the sinking of the "Titanic" is, whether or not a ship could sink to the bottom of the sea in a depth of 2000 fathoms. A short time before that disaster I presented a paper before the combined Civil Engineering and Mechanical Engineering Societies of the Massachusetts Institute of Technology, from which paper I will make some extracts, thinking that the subject might be of interest to POWER readers.

"Water is reduced in bulk by 1 per cent. when under a pressure of 3000 lb. per square inch, and this reduction increases in direct proportion to the pressure, so that there is a reduction of 10 per cent. at a pressure of 30,000 lb. per square inch.

"Applying this value to the density of the ocean at its greatest depths gives this compression in a different form. There is a depth of 31,614 ft. (5.9 miles) in the Western Pacific Ocean, where the pressure would equal 13,752 lb. per square inch, or 942 atmospheres; the weight of a cubic foot of sea water would there be increased from 64 lb. at the surface to about 67 lb. at the bottom. This increased density of the ocean in the lowest depths does not, however, prevent the sinking to the very bottom of anything that once starts in that direction.

"There is a popular idea that submerged objects continue to sink only until they reach a certain pressure, which is supposed to have some relation to their specific gravity, when they will sink no lower, but will be carried by undersurface currents here and there, until they finally reach some Sargasso Sea of wreckage. It may be safely assumed, however, that anything that will sink in a glass of water will sink to the bottom of the sea, for, as the depth and the pressure increase, the sinking object is subjected to so intense a pressure externally that compression and even implosion occurs, and it is reduced to so small a bulk that its specific gravity is greatly increased."

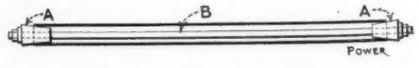
JAMES W. NELSON.

New York City.

Unnecessary Risks

The editorial "Unnecessary Risks," in the Apr. 30 issue, mentions entering ashpits under a bright fire, which brings to mind a task that was occasionally performed on board ships a few years ago; only in this case it was a necessary risk.

When a boiler tube gave out before the patented tube stoppers came into use, a sure and quick repair was necessary. The illustration shows the method of plugging a Scotch marine boiler tube.



PLUGGED BOILER TUBE

Pinewood plugs *A*, tapered to easily enter the tube ends, were carried and one plug had to be put in from the combustion-chamber end. The plugs were bored to receive a rod *B* threaded at each end and nuts and washers secured both plugs in place, thus cutting out the leaking tube and making it possible to keep the boiler in service until a new tube could be put in.

It was quite easy to place the plug at the front end, but not so at the back end, for the engineer had to enter the furnace under the nest of tubes, in which the leak was located, to put in the plug. The fire was smothered with green fuel, and placing a plank laid on top of it, and

crawling on his hands and knees, a pine plug in one pocket and a nut and washer in the other, the engineer entered the combustion chamber. He quickly placed the plug on the rod, which was already pushed through from the front end, and tightened the nut and plug in place, the final tightening being done from the front end.

On again entering the boiler room the engineer was covered with a blanket to prevent a severe chill. Sometimes after coming from a hot boiler furnace a man faints when he strikes the cooler air of the fire room, even though it may be as high as 115 deg.

The success of the job lies in keeping one's head long enough to enter the plug and screw on the nut. It does not take many seconds to do it if proper preparation is made, but everything must be just right, and there must be no bungling with either the plug or nut, for the heat is intense and more or less steam and water are spattering over the operator; for this reason an engineer performs the job.

CHARLES J. MASON.

Scranton, Penn.

Boilers Transfer Water

I once had boilers transfer water as described by Mr. Warner in the issue of Apr. 9. The plant contained four return-tubular boilers of 200 hp. each. the feed pipes entering through the top and extending down between the tubes below the water line, terminating in a tee with a piece of perforated pipe screwed in each end.

It seemed to me that No. 1 boiler was a rapid steamer, for immediately after the fire was cleaned in No. 1 furnace the stop valve on the other boilers would require partial closing to keep water in No. 1 with the pump working, and if the pump was stopped the water would go down in No. 1 and rise in the other boilers. The check valve appeared to be all right, but I took it out and replaced it with a new one. After that there was no more transfer of water.

The cause of the trouble is obvious: No. 1, being hotter than the others, owing to its clean fire, carried a slightly higher steam pressure which forced the water up the feed pipe, through a defective check valve and into the other boilers. I would advise Mr. Warner to keep the feed pump going with water near the middle gage and grind or replace the defective check valve.

N. R. WATKINS.

Merritt, B. C.

Answering W. E. Warner's problem under the above title, in the Apr. 9 issue, I think he will find a leaky check valve on the boiler-feed line leading to No. 2 boiler. It is impossible for the water to go over the U-bend back into No. 1 even if No. 2 primes, especially if either engine is in operation. Mr. Warner does not show a diagram of the feed line from the pump, but his trouble must be there. I had the same thing happen on a pair of boilers I was once operating, and after studying the matter for some time, I put in a new check valve and the trouble vanished.

The "other engineer" is mistaken in thinking No. 1 is the better steamer; the reverse is true. If No. 1 was the better steamer or the hotter, the water would not flow from No. 2 to No. 1 because the pressure would be a little greater in No. 1 and hold the water back. As it is, No. 2 is the easier steamer and is a little hotter, so it overcomes No. 1's pressure and sends the water through the defective check valve. The more water No. 1 gets, the cooler it becomes and the less its pressure. If there is no other connection, the trouble must be in the feed line.

There is a battery of two boilers connected similar to Mr. Warner's in our plant. The safety valves are set exactly alike; sometimes No. 1 will blow, sometimes No. 2. One of the firemen asked me why it was that "No. 2 always blows for him while No. 1 always blows for John."

I asked him which boiler he fired first and he said No. 1, and John always fired No. 2 first. One fireman was a right-hand and the other a left-hand shoveler. I explained that the hotter boiler would blow off first, the valves being set exactly alike, and No. 2 blew for the left-hand shoveler because he fired No. 1 first and invariably when he got to No. 2 he would think the boilers would stand a little more and would send in half a shovel of coal more into No. 2 than he had in No. 1.

I proved it to him in less than two hours by throwing half a shovelful more into No. 1 than No. 2; No. 1 blew the valve. The next day he unconsciously fired his old way and blew No. 2. I require the safety to be blown at least once in every 24 hours, not oftener if it can be helped, yet I do not complain if it blows twice.

C. A. SCOTT.

Wales, Wis.

A few years ago I was acting as water tender in a plant where the boilers were all connected to one header and were forced to their limit. We were using forced draft and great care was required to fire regularly and evenly, to avoid the trouble Mr. Warner mentions.

The chief engineer said that it was due to water being transferred from one boiler to the others. I never tried to prove it, and, like Mr. Warner, would like to know the cause of the trouble.

HAROLD J. WILKINSON.

Angola, Ind.

[The blowing of the high-water alarm mentioned by Mr. Warner is not due to a transfer of water from one boiler to the other. The higher water level, which causes the alarm to blow, is produced by a more energetic evaporation of the water in the boiler in which the alarm sounds.—EDITOR.]

Manila and Cotton Rope Transmission

My experience with rope drives may be of interest. In a cement mill, several of the departments were driven by three-cylinder vertical gas engines of about 280 hp. at 200 r.p.m. Each engine drove a line shaft through manila ropes with 9-ft. pulleys at about 40-ft. centers. The American or continuous system was used.

The ropes gave a great deal of trouble by swaying, especially if one of the cylinders was not acting properly, causing the engine to give a jerky motion to the rope. This swaying was so bad that flumes were built along the ropeways to keep the side-slapping within limits, and prevent the ropes from leaving the wheels, but the flumes kept them in place only fairly well as occasionally a rope would leave the pulley, get mixed up with the hub and shaft; then a new rope was necessary. This happened about nine times a year in the most troublesome drives.

Finally cotton ropes were tried with results that were highly satisfactory. The swaying almost entirely ceased, and one of the most troublesome drives now ran so quietly that the ropes resembled pipe lines.

E. D. GAGNIER.

Salem, Ohio.

Forces of Expansion and Contraction

On page 351 of the Mar. 12 issue, Mr. Ingham states that a gland joint inserted for expansion in a steam-pipe range between two boilers was blown out owing to the anchor bolts being improperly fitted and that this caused the boiler to be moved on its setting.

I am at present engaged upon an arrangement of steam piping and should be glad if Mr. Ingham, through POWER, would favor me with a sketch of the piping arrangement where this occurred as I have no wish to undergo a similar experience.

D. VARLEN.

London, N., England.

The instance mentioned in my article on the "Forces of Expansion and Contraction" in the issue of Mar. 12, was described in detail in a leading English engineering journal some years ago. Un-

fortunately, I am unable to find the description or illustrations. If, however, Mr. Varlen will take care to fit guard bolts to any gland-expansion joints he inserts in the pipe range, he need have no fear of undergoing a similar experience to the one referred to.

EDWARD INGHAM.

Manchester, England.

What the Diagrams Show

In the Mar. 26 issue J. F. Reynolds submitted a set of indicator diagrams taken from a compound pumping engine. He asked for a criticism on the receiver pressure carried and the distribution of work in the high- and low-pressure cylinders.

At first, the diagrams look excellent and the distribution of work nearly equal, and this would be the case if it were not for the high ratio of cylinder areas. The following calculation of horsepower delivered shows that the low-pressure cylinder is doing more than twice the work done in the high-pressure cylinder:

High-pressure cylinder

$$I.hp. = \frac{34 \times 2.5 \times 113 \times 30 \times 2}{33{,}000} = 17.5$$

Low-pressure cylinder

$$I.hp. = \frac{11.2 \times 2.5 \times 707 \times 30 \times 2}{33{,}000} = 36$$

making a total of 53.5 hp. Then,

$$\frac{36}{53.5} = 0.67 \textit{ per cent.}$$

Thus the low-pressure cylinder is doing 67 per cent. of the work. If the cylinder ratio were 1 to 4, with the same size high-pressure cylinder and the same mean effective pressures, then the horsepower delivered would be 17.5 in the high-pressure cylinder and 23 in the low-pressure. This would mean a distribution of 57 and 43 per cent. respectively.

The ratio of areas is actually 1 to 6¼. Therefore to obtain equal work in the two cylinders the mean effective pressure of the high-pressure cylinder should be 6¼ times that of the low-pressure. This would mean that with the same total load as stated, the high-pressure cylinder mean effective pressure should be about 51.5 lb. and that in the low-pressure 8.25 lb. To bring this about, the high-pressure cylinder should cut off later and the low-pressure earlier, with a resulting receiver pressure of about 20 lb. gage.

If the engine were loaded up to rating, it is likely that the engine would automatically adjust itself to a more equal distribution of the load; the high-pressure cylinder would cut off later with a resulting increase in power; the receiver pressure would be higher, and the power delivered by the low-pressure cylinder would increase, but not in proportion to that of the high-pressure.

Mr. Reynolds remarks that, although there is very little compression, the engine runs smoothly. This is undoubtedly due to the slow speed of the engine and the effect of the water cylinders if they are connected tandem to the steam cylinders.

I notice also that on the crank end of the high-pressure cylinder, the release occurs at about ⅞ stroke. With the present point of cutoff the effect is slight, but with a later cutoff it might be well to improve this function.

LOUIS L. HEBBERD.

Chicago, Ill.

The volumetric ratio of the cylinders of this engine is

$$\frac{302}{122} = \frac{6.25}{1}$$

Therefore, for equal work the mean effective pressure of the low-pressure cylinder multiplied by 6.25 should equal that of the high-pressure cylinder. In this case, the average mean effective pressure of the high-pressure is 34.125 lb. and of the low-pressure 11.25 lb.; the ratio of work is

$$\frac{11.25 \times 6.25}{34.125} = 2.06$$

or the low-pressure cylinder performs 2.06 times as much work as the high-pressure cylinder.

If the engine is arranged with the cylinders in tandem, the distribution of work is of small consequence if the economy is good, but if it is a cross-compound, with its cranks at 90 deg. apart, the unequal work may cause an objectional and serious disturbance of the crank velocity.

If equal work is desired, it can be had by the following process. The indicated work shown is 53.69 hp.; thefore the mean effective pressure (total) referred to the low-pressure cylinder must be

$$\frac{53.69 \times 33{,}000}{706.86 \times 150} = 16.71 \textit{ lb.}$$

To do one-half the work in the low-pressure cylinder, its mean effective pressure must be one-half of 16.71, or 8.355 lb., and for equal work in the high-pressure cylinder its mean effective pressure must be

$$8.355 \textit{ lb.} \times 6.25 = 52.218 \textit{ lb.}$$

The equalization may be had by adjusting the cutoff by means of the knock-off cams until the correct mean effective pressures are obtained in the diagrams.

Mr. Reynolds' comment relative to smoothness of operation with the small compression shown requires no comment except that, were it not for the danger of blowing through, a still further sacrifice might be advisable as it is well known that the resistance due to the head of water against which the engine works is usually ample for the cushioning of the reciprocating parts of the machine.

CHARLES F. PRESCOTT.

Philadelphia, Penn.

Mine Pump Blew Out Gasket

In the Apr. 2 issue, the sketch accompanying Mr. Saxe's letter does not agree with my experience with mine pumps. I have found that the relief valve should be closer to the pump than the check valve.

A mine pump similar to that under discussion is on the bottom of a 400-ft. shaft. The brass disk in the check valve was fastened to the arm with a bolt and nut. The nut came off, loosening the disk and wedging it in the end of the check valve, stopping the flow. If the valves had been arranged as in Mr. Saxe's sketch, there would have been a cracked air chamber or something more serious. The relief valve saved the pump and showed us where the trouble was located.

L. C. MOORE.

Ishpeming, Mich.

Breakage of Separator Gage-Glasses

Replying to H. L. Burns' letter under the above heading in the Apr. 16 issue, it is generally conceded that sudden temperature variations are the cause of such breakage, but I would like to know what theory Mr. Burns offers on his idea. I have considerable trouble and annoyance with gage-glasses breaking and will try out his suggestion, but I would like to know the fundamental reasons why the pipe assists in lengthening the life of the glass.

JOHN KLEMM.

Mexico.

A Troublesome Feed Pump

Replying to A. J. Berlin's letter under the above title in the Apr. 12 issue, my feed pump failing to supply enough water to the boiler, I opened the petcock on the water-end and was surprised to find a full boiler pressure. After closing the pump throttle valve and relieving the feed line of pressure, I removed the cap of the check valve at the boiler. It was found that the nut which holds the disk in place had unscrewed almost entirely from the stem.

The nut had three projections on its lower side, which held the disk while regrinding the valve. As the nut had become loose and the valve raised to allow the passage of water, the projections acted as impeller blades propelled by the water, thus unscrewing the nut until it rested on the bottom of the valve, preventing it from closing and the pump from doing its work. After tightening the nut and spreading the stem with a center punch, the valve was reassembled, and the pump ran without further trouble.

A. R. KUMMERER.

Hazleton, Penn.

Inquiries of General Interest

All Questions Must be Accompanied by Name and Address—Not for Publication

Storing Bituminous Coal

What is the best method of guarding against spontaneous combustion of bituminous coal stored in a bunker?

H. B. L.

The bunker should not be filled over 10 to 12 ft. deep. Spontaneous combustion may be guarded against by any procedure that will keep the mass of coal cool. One means is to have the bottom of the bunker of plate iron cooled by natural or induced draft of air. Another is to have vent ducts near the bottom of the bunker with shields or covers arranged so that cold air may enter the vent ducts through the sides or bottom of the bunker and circulate up through the mass of coal.

Suitable Size of Feed Pump

What size feed pump would be suitable for a 1000-hp. boiler? How may the fire baffles of a B. & W. boiler be inspected?

L. A. S.

A 10x7x12-in. single or 8x5x12-in. duplex boiler feed pump would be suitable for a 1000-hp. boiler. The fire baffles of a B. & W. boiler may be inspected for leakage while the boiler is in use by opening one side dust door after another, and observing whether the flame passes through the baffles. When the boiler is not in use the baffles may be inspected by removing the top covering of the boiler and dropping a light down between the tubes.

Use of Exhaust Condensation as Feed Water

Can condensation from exhaust steam heating be employed as feed water without danger of burning boiler shell due to deposit of oil in boiler?

J. D. M.

It is very difficult to remove all traces of cylinder lubricating oil from the condensation, and when the feed water consists largely of condensation from exhaust, the latter is likely to deposit oil in the boilers, the amount depending on the combination of such conditions as kind and quantity of oil used in cylinders of engines, analysis of the original feed water, activity of boiler circulation, intensity of firing, etc. In many plants all of the condensation of exhaust that is recoverable is used as feed water without serious injury to the boilers, in most instances employing some method of removing oil from the exhaust steam or from condensation or both. But in any instance where exhaust from condensation has never been tried as feed water, it should be used sparingly at first and gradually increased as a proportion of the total feed water required, with frequent internal examinations of boilers for discovery of oil deposits.

Power Developed by Pumping Engine

How many indicated horsepower should be developed by a triple-expansion duplex direct-acting pump, each side having high-pressure cylinder 10 in. diameter, intermediate 16 in. diameter and low pressure 30 in. diameter by 24 in. stroke, supplied with steam at 125 lb. boiler pressure and operated condensing with a 26-in. vacuum and making 55 strokes per minute?

B. E. T.

Supplied with dry steam the pumping engine should develop 147 i.hp.

Siphons or Jet Pumps

What difference if any is necessary in proportions of nozzles in siphons to work by water in place of steam pressure?

E. G. H.

The main difference is that in jet pumps which are operated by water pressure, the passages for the pressure supply are larger in cross-sectional area and the reductions are usually made more abrupt.

Degree of Vacuum in Heating Coils

When a vacuum gage placed on the return receiver of a vacuum-exhaust steam-heating system indicates 20 in. vacuum, will there be the same vacuum in the heating coils?

T. B.

The vacuum in coils can never be the same as that of the receiver except while a coil, or radiator too small for its work, acting as a condenser, is becoming filled with water.

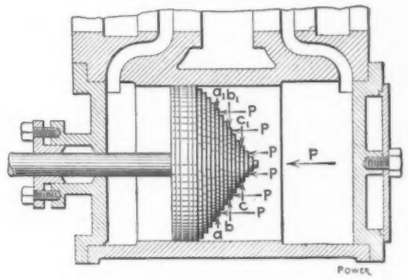
Pressure on Conical Piston

When the piston of an engine has a conical or curved surface which exposes a greater area to the steam than if it had a plane surface, why is it that in estimating the power developed, the piston area is treated as though it were a plane surface at right angles with the stroke of the piston?

F. B.

Steam within the cylinder of an engine exerts equal pressure on every unit of area of the exposed surface of the sides and head of the cylinder and of the piston. Regardless of the shape of surface thus exposed to pressure, the resulting total pressure exerted on any particular area, in any particular direction, is the product of the pressure per square inch by the area in square inches of the surface projected on a plane at right angles to the direction of pressure under consideration. Hence, the total force exerted on a piston by the steam in the direction of the stroke, is the pressure of steam per square inch multiplied by cross-sectional area of the cylinder in square inches.

A graphical analysis of the pressure of steam on a conical-shaped piston is given in the illustration. The conical surface may be considered to be made up of infinitely thin disks, similar to those of the drawing, increasing uniformly in diameter toward the base of the cone and the following would still hold true. The total pressure resolved in direction *P, P, P,* is effective on the exposed vertical surfaces of the disks, the total area of which is equal to the cross-sectional area of the cylinder. Any radial pressure like *a, b* or *c* against the edges of the disks is counteracted by equal and opposite pressure, as at a_1 b_1 c_1. Similarly, a piston of any other form may be resolved into equivalent surfaces parallel and perpendicular to the axis of the cylinder, and the effective area will always equal the cylinder's cross-sectional area.



PRESSURE ON CONICAL PISTON

The Phipps power plant, which has for years been furnishing light, heat and power to a group of Pittsburgh buildings, is now owned by the Duquesne Light Co.

Study Questions

This Week's Questions
Last Week's Answers

16. With the fewest cuts and welds, how can six pieces of chain be joined into one piece if two pieces contain two links; two, three links, and two, four links?

17. An engineer had a trench to cover, 12 ft. long by 2 ft. wide. He happened to have on hand a board 3 ft. wide and 8 ft. long. How could he completely cover the trench by cutting this board into only two pieces?

18. What will be the energy in foot-pounds in a blow from a steam hammer if the weight of the moving parts is 62 lb., length of stroke 8 in., diameter of cylinder 6 in. and mean effective pressure 90 lb. per square inch, neglecting friction and atmospheric resistance?

19. Having obtained the energy in problem 18, find the velocity of the hammer at the end of the stroke.

20. A cylindrical boiler shell is 66 in. in diameter by 16 ft. long. With 90 lb. gage pressure what is the total force tending to rupture the shell along one longitudinal seam?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

11. The volume of the engine foundation can be found by the prismoidal formula.

$$V = \frac{h}{6}(A_1 + 4\ A_m + A_2)$$

where

V = Volume;
h = Altitude;
A_1, A_2 and A_m = Top, bottom and middle areas respectively.

The problem gave

$h = 2$ ft.;
$A_1 = 2 \times 4 = 8$ ft.;
$A_2 = 3 \times 5 = 15$ ft.

The middle area is found by multiplying the means of the top and bottom lengths and widths.

$$\frac{2+3}{2} = 2\tfrac{1}{2} \qquad \frac{4+5}{2} = 4\tfrac{1}{2}$$

$$\frac{5}{2} \times \frac{9}{2} = \frac{45}{4} = 11\tfrac{1}{4}\ sq.ft.$$

Substituting in the formula,

$$V = \frac{2}{6}(8 + 45 + 15) = \frac{68}{3} = 22\tfrac{2}{3}\ cu.ft.$$

Multiplying by 125 lb., the weight of the foundation is 2832.5 lb. Adding the weight of the engine and dividing by the area of the base,

$$\frac{2832.5 + 650}{3 \times 5} = 232.16\ lb. =$$

the load per square foot on the floor.

12. Let x = the number of tons burned in April. Then $x \times \frac{x}{100}$ = gain in May = $56 - x$. Simplifying the equation,

$$\frac{x^2}{100} + x = 56$$

or

$$x^2 + 100x = 5600$$

This is a quadratic equation requiring for its solution the completion of a perfect square in the left-hand member, which is done by adding the square of one-half of the coefficient of the second term to each side of the equation, or

$$x^2 + 100x + \frac{(100)^2}{2} = 5600 + \frac{(100)^2}{2}$$

$$x^2 + 100x + 2500 = 8100$$

Extracting the square root of both sides,

$$x + 50 = 90 \text{ or } x = 40\ tons.$$

This proves since 40 per cent. of 40 tons is 16 tons, the gain in May over April.

13. As during the first hour he would be carried west a distance of two miles, in traveling his first three miles he must take such a course toward the northeast as to arrive at a point which would be the apex of a right-angle triangle having a base of two miles, a hypothenuse three miles and consequently a perpendicular 2.236 miles long. As each hour of travel in the direction of the hypothenuse carries him 2.236 miles, then to be carried five miles would require

$$\frac{5}{2.236} = 2.236\ hr. = 2\ hr.\ 14\ min.\ and\ 9.6\ sec.$$

14. All of the forces acting on the ladder can be considered as acting about the base as a hinge and their moments (the products of the forces by their arms, which are the perpendicular distances between the lines of direction of the forces and the hinge center) must add up to zero. Two of the forces are vertical, the weight of the ladder and the weight of the man; the reaction of the wall is horizontal. The weight of the ladder will act through its center of gravity, which, since the ladder is of uniform weight, is 15 ft. from the end. The arm of this force will then be

$$15 \times cos.\ 65\ deg\ 50\ min.$$

From a table of natural trigonometrical functions

$$cos.\ 65\ deg.\ 50\ min. = 0.40939$$

Then

$$15 \times 0.40939 = 6.14085$$

and the moment of the weight will be

$$62 \times 6.14085 = 380.7327 \qquad (a)$$

Similarly the arm of the force corresponding to the man's weight at 25 ft. from the foot of the ladder is

$$25 \times 0.40939 = 10.23475$$

and the moment

$$150 \times 10.23475 = 1535.2125 \qquad (b)$$

The arm of the reaction of the wall is the vertical height of the point of contact of the ladder from the ground, or

$$25 \times sin.\ 65\ deg.\ 50\ min. = 25 \times 0.91236 = 22.809$$

and the moment

$$X \times 22.809 \qquad (c)$$

This reaction being opposed to the other forces

$$(c) - (a) - (b) = 0$$

or

$$(X \times 22.809) - 380.737 - 1535.2125 = 0$$

$$X = \frac{1915.9452}{22.809} = 83.9995\ lb.$$

And as action and reaction are equal, the pressure of the ladder against the wall is the same or approximately 84 lb.

15. Let x = speed of slower pulley. Then $x + 100$ = speed of faster pulley and

$$x + x + 100 = 110$$
$$2x = 10$$
$$x = 5$$

The speeds of the two pulleys are therefore 5 and 105 r.p.m.

Illinois N. A. S. E. Decatur Convention

A large number of delegates and friends attended the Illinois National Association of Stationary Engineers' eighth annual convention at Decatur, on May 23, 24 and 25.

Thursday's Session

The first session was opened on Thursday morning, by an invocation by Rev. W. H. Penhallegon, after which Mayor Dan F. Dinneen delivered a brief address of welcome, in which he touched upon the importance of the engineer in the community. He likened the power plant to the mainspring of a watch; when that fails everything else becomes idle and useless. National Secretary Fred Raven responded briefly on behalf of the association. Judge O. W. Smith spoke of the layman's ignorance of the engineer and his work.

National Vice-president John McGrath contrasted the high development of steam engineering with that of 30 years ago, and expressed the opinion that, although marvelous strides have been made, this is only the beginning of the progress yet to be made.

John Lane, editor of the *National Engineer*, said that the vast amount of reckless waste that has existed, and still does in some industries, was characteristic of the pioneer, but keen business competition and other causes are eliminating much of it. One of the main functions of the N. A. S. E. is to educate the engineer to conserve the raw materials with which he deals.

W. L. Osborne, secretary-treasurer of the Central States Exhibitors Associa-

tion, speaking for President F. E. Ramsley, who was unable to be present, told of the objects of the exhibitors and their association. State Deputy W. L. Parker expressed his pleasure in the healthy condition of the organization and the bright future in store for it, and Past National President Beckerleg told of the life- and accident-insurance department and the good it is accomplishing.

Local President E. R. Wortman then turned the gavel over to State President J. F. Alt, who appointed the various committees and declared an adjournment until 2 p.m.

At the afternoon session a proposed license law, prepared by a committee appointed last year, was taken up and approved with a few minor modifications. The law provides for a chief examiner and not less than eight district examiners who shall issue licenses to applicants after suitable examination. Anyone who wishes to operate a boiler carrying a pressure of 15 lb. per square inch or more, or a steam engine, or other piece of steam-driven apparatus of 15 hp. or more rated capacity, must first obtain a license.

At 3:30 the session was adjourned to permit a visit to the works of the H. Mueller Manufacturing Co., maker of plumbing faucets and valves, pressure regulators, etc.; also to the Decatur Ry. & Light Co. and the Decatur Brewing Co. plants.

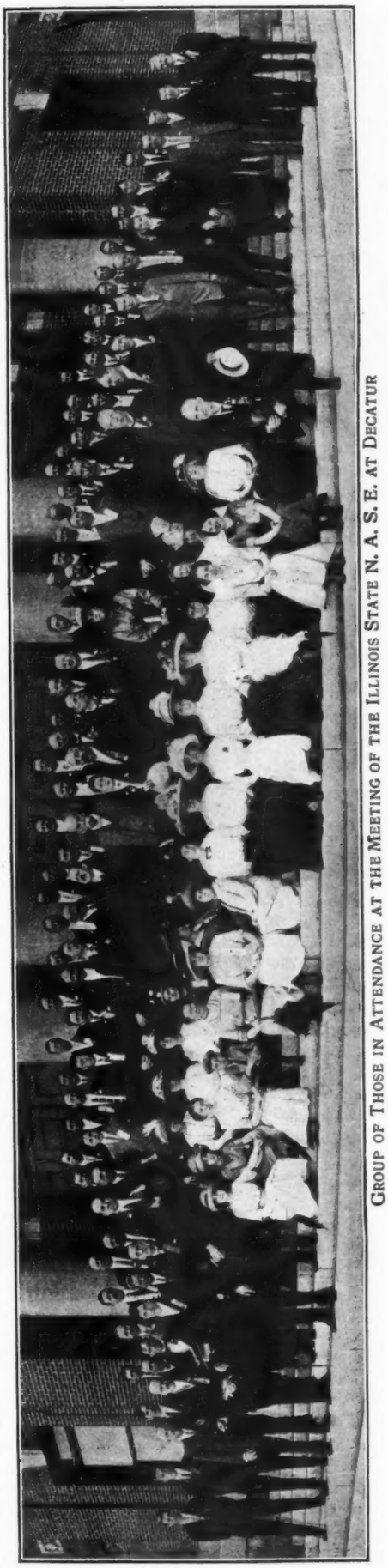
At the evening session, Mr. Freeman, of the H. Mueller Manufacturing Co., read a paper prepared by A. W. Cash, of the same company, on "Steam Pressure Reducing Problems," in which it was pointed out that the trouble with many of the reducing-valve installations was that too large a valve was employed. Where the steam pressure is reduced it is best to place the reducing valve as far from the engine as possible, thus reducing the effect of the pulsations caused by the opening and closing of the admission valves.

Fred Hickey, of the Dearborn Drug & Chemical Co., then told humorous stories and gave some laughable imitations.

Friday's Session

On Friday the delegates and friends took a 50-mile trip to Urbana and visited the University of Illinois. Dean Goss welcomed them and expressed the hope that the university and the engineers throughout the state might become more closely associated to the greater advantage of both.

Prof. S. W. Parr, one of the foremost fuel experts of the country, gave a short lecture on "Specifications for the Purchase of Coal." He showed the necessity of specifications where in many locations coals of widely varying compositions and fuel worth are available, though in many cases the specifications



Group of Those in Attendance at the Meeting of the Illinois State N. A. S. E. at Decatur

are unnecessarily complicated. The important thing is heat value and it is possible to draw simple specifications to give the purchaser reasonable protection.

Prof. Edward Bartow, director of the Illinois State Water Survey, gave a short discourse on "The Work of the State Water Survey with Special Reference to Boiler Water." The survey makes free analyses of water for sanitary purposes. It has analyzed a great number of waters for mineral content and has issued figures on the cost of softening them. Prof. Bartow showed how important it is to use a boiler-feed water which will not form scale, by stating that $\frac{1}{16}$ in. of scale on the heating surface means a fuel loss of 15 or 16 per cent., not to mention the repair costs made necessary by burned tubes, plates, etc. The visitors then inspected the fuel-testing and water-supply laboratories.

After luncheon in the Y. M. C. A. café the party assembled in the engineering lecture room where J. Paul Clayton gave a lecture on "A New Analysis of Indicator Diagrams," illustrated with lantern slides. By replotting a diagram on logarithmic paper it is possible, said the speaker, to ascertain the volumetric clearance very accurately, to estimate closely the steam consumption and to discover the presence of leaks as well as check up other important points as to the condition and operation of the engine. The university will shortly issue a bulletin fully covering this subject.

Prof. Leutwiler explained the method used in teaching power-plant designing. Each student is given a specific problem in which the controlling factors, such as the load curve, location, cost of fuel, etc., are given and he is required to lay out a suitable plant, make a general drawing and a piping plan.

Adjournment was then made to the students' drafting room, where some of the actual work was examined. In groups of half a dozen and under the guidance of the professors and some of the senior students, the party made a tour of the laboratories of mechanical engineering, electrical engineering and materials testing, the power house, blacksmith shop, machine shop, foundry and pattern shop. The visitors then returned to Decatur.

Saturday's Session

On Saturday morning the delegates assembled and after Rockford was selected as the place of the next annual convention, the following officers were elected: President, John Morgan, Association No. 3, Waukegan; vice-president, G. W. Hullinger, Association No. 16, Decatur; secretary-treasurer, W. E. Hill, Association No. 17, Rock Island, reëlected.

The new officers were installed by National Secretary Raven, and J. F. Farrell, ex-mayor of Ottawa, made an address

in which he strongly sympathized with the engineers and appreciated the obstacles which they encounter. He thought that it would be beneficial to have more public meetings so that the plant manager and owner might see what the engineers are doing, what they need and deserve, and would do all in his power to aid the Illinois State Association to bring about the enactment of the license law advocated by it.

Prof. E. C. Woodruff, of the James Millikin University, then talked briefly on higher education for engineers, after which the convention was adjourned.

The Exhibits

The exhibition hall, under the charge of the Central States Exhibitors Association, was well arranged and numerous effective displays were made. Those exhibiting were American Steam Gauge & Valve Manufacturing Co., V. D. Anderson Co., Crandall Packing Co., Decatur Railway & Light Co., C. A. Elliott, representing the George B. Carpenter Co., Dearborn Drug & Chemical Co., Faries Mfg. Co., Field & Shorb Co., Garlock Packing Co., Greene, Tweed & Co., Hawk-Eye Compound Co., Hills, McCanna Co., Home Rubber Co., Jenkins Bros., W. H. Jennings, H. W. Johns-Manville Co., Keystone Lubricating Co., Lunkenheimer Co., H. Mueller Co., Osborne Valve & Joint Co., William Powell Co., Power, Ralph H. Rawson, representing Anchor Packing Co., Maguire Rubber Co. and McMaster-Carr Supply Co., Schaeffer & Budenberg Manufacturing Co., Standard Oil Co., C. E. Squires Co., Wickes Boiler Co. and the United States Graphite Co.

Exhibit at Pratt Institute

The annual exhibit of the work of day students of Pratt Institute, Brooklyn, N. Y., will be held on Saturday, June 15, 10 a.m. to 5 p.m.; Monday, June 17, 10 a.m. to 10 p.m.; Tuesday, June 18, 10 a.m. to 5 p.m. This exhibit is public and affords those interested in industrial and technical education an opportunity not only of viewing the students' work in the various courses, but of inspecting the methods, the equipment and the general facilities of the school for conducting this training.

There are day and evening courses for both men and women in a great variety of vocational subjects, including fine and applied arts, architectural design and construction, household science and arts, library management, kindergarten training and technical and trade courses for men along several important lines.

The work of the School of Science and Technology is of especial interest to men engaged in technical and trade pursuits. This school provides instruction in applied mechanics, applied electricity, applied chemistry and tanning, machine work and tool making, carpentry and building and pattern making.

The twenty-fifth anniversary of the founding of the Institute will be celebrated with a dinner, June 15, to which all graduates and former students are invited.

Standards for Fittings and Valves Not Settled

A warning is being sent out by the committee of manufacturers on Standardization of Fittings and Valves, 30 Church St., New York City, not to specify or order flange fittings by the 1912 U. S. Standard for standard weight and extra-heavy flanges and flange fittings, adopted by the National Association of Master Steam & Hot Water Fitters. This standard has not been accepted as final by the manufacturers of flanged fittings and valves, and they state will not be accepted until certain necessary revisions have been made to suit the practical requirements of all concerned.

Before a new standard which deviates from the generally accepted practice for flanges and flanged fittings can be considered final, a number of important points must be carefully considered. In view of these considerations it is better to maintain the old practice until a universal standard can be agreed upon.

OBITUARY

Wilbur E. Richardson

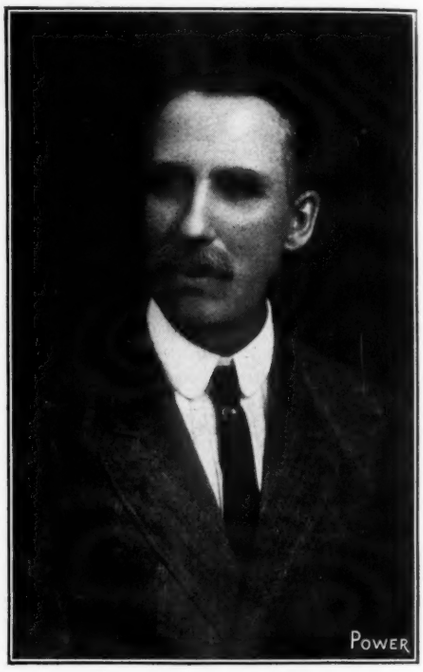
On May 15, Wilbur E. Richardson, president and general manager of the Richardson-Phenix Co., Milwaukee, Wis., died of apoplexy at his late home, 875 Island Ave., Milwaukee.

Early in life, Mr. Richardson served his time as machinist in the Marionette Iron Works, Marionette, Wis. He early demonstrated his ability to cope with difficult problems in general power-plant work, and soon accepted the position of chief engineer of the Gilbert Paper Co., Menasha, Wis. The improvements and economies which he effected in this plant brought him into prominence and he was soon induced to enter the employ of the E. P. Allis Co., at Milwaukee, where he established an excellent record in this company's erecting and testing departments.

Later, while engaged in erecting pumping engines, Mr. Richardson conceived the idea of developing a mechanical cylinder-oil lubricator along new lines. He submitted his scheme to Edwin Reynolds, of the Allis-Chalmers Co., who thought so well of it that he gave Mr. Richardson his first order. The business was then established in a small way in Milwaukee.

In the fall of 1900, Mr. Richardson was sent to New York City to complete the installation of eleven 5000-hp. engine units in the Ninety-sixth St. station of the Metropolitan Street Railway Co. His successful work at this plant led to his promotion as the Allis-Chalmers Co.'s superintendent of erection for the Eastern district. Mr. Richardson took charge of similar large installations in the Manhattan Ry. Co.'s Seventy-fourth St. power station, the Interborough Rapid Transit Co.'s Fifty-ninth St. station and the Brooklyn Rapid Transit Co.'s Third Ave. power station. Upon the completion of these installations, Mr. Richardson was appointed general superintendent of all the work of the Westinghouse Machine Co.

In the meantime, the mechanical lubricator business which Mr. Richardson had established several years before had



W. E. Richardson

grown to such an extent that in 1907 he was enabled to sever his connection with the Westinghouse Co. and devote all of his time and energy to the design and manufacture of his company's products.

Mr. Richardson's inventive genius soon developed a number of important improvements for the economical application of oils to power-plant machinery, many of which are used today in the large power plants throughout the country.

His foresight and excellent judgment caused him to surround himself with a business organization that is in such excellent shape today that his sudden taking away, while a great shock to his associates, will not in any way interrupt the business.

Mr. Richardson was endowed with a strong and genial personality which had gained for him wide and steadfast friendships throughout the country. He was a keen judge of human nature, and his memory will be particularly cherished by many engineers whom he had placed in good positions. He is survived by Mrs. Richardson and two sons.

Moments with the Ad. Editor

Advertising "works in devious ways its wonders to perform." Some folks go at it single-purposed—they realize that things are to be learned from advertisements.

Others, like the millionaire's baby, have riches thrust upon them. Still others pick up a well-stuffed pocket book when the policeman is at the other end of his beat.

The first named class get the most out of advertising.

We advocate "consistency" in reading the Selling Section of POWER.

You know that only truthful advertising pays.

You know that we guard these pages carefully and keep the crook out of them.

Therefore, you can read the Selling Section with a full measure of faith in what's printed there.

How well it paid one engineer is plainly shown in the incident contributed here by C. A. Tupper, the well known consulting engineer of Milwaukee, Wis.

A BETTER ENGINEER

By C. A. Tupper

Some weeks ago I was visiting a large manufacturing plant, when the manager suggested, before I left, that I take a look at their power house in which they had considerable pride. This contained five engine driven electric and compressor units. It was well arranged, and everything about it seemed more than ordinarily up to date.

Observing an elevated tank in one corner of the station, above the engines, and afterwards an oil filter in the basement, I spoke about the latter.

"Yes," said the engineer in charge, "that has been quite a saver for us. We are now using 4 barrels of oil for external lubrication per year, as against 12 to 15 barrels before we put in the filter system."

"How did you happen to install it?" I asked.

"Well, that's a funny thing," he replied. "Of course I realized the advantages of filtering, and we've had any number of circulars, letters and folders on automatic filters from different concerns; but there are always so many things I have to requisition for that I hated to recommend going to the expense of putting in the system. Advertisements on that particular subject I had never paid much attention to.

"One day, however, after I had been reading some of the operating kinks in POWER, I was turning over the advertising pages, sort of casually, when I was attracted by a statement from the ——— people that one of their filters in a Western plant had cut the bill for lubricating oil down to less than 40% of what it had been. They gave the actual figures for six months' operation and stated that this was an average example; that they would be glad to tell anyone interested about other specific cases, some of which showed considerably greater saving.

" 'Great Scott!' said I to myself, 'If we can make as much of a saving as that, it won't take long, at the present price of oil, to pay for adding an automatic filter to our plant;' so I took the matter up with our manager, Mr. Weir, and he telephoned their local agent to come out and see us. The figures that he showed, with letters from users, made it easy to convince us and we had the filter installed on trial. The result was that we paid for it within discount time."

"Hadn't the local agent of the filter manufacturers ever called on you before?" I inquired. "It seems to me that he wasn't very enterprising."

"Oh, yes, he had called several times," the engineer answered, "but once or twice he didn't get past Mr. Weir, another time he struck me when we were having trouble in our basement on account of high water in the river, and once he came just after I had been told that we positively must keep down any but the most necessary expenses; so he really was not listened to. I suppose there had been good ads in the paper before, too; but this one happened to bob up just at the right time."

"Haven't you had similar experiences with new apparatus, or even improvements in some standard lines?" I queried.

"Oh, yes, lots of times. I'll see a thing advertised week after week without more than just glancing at it, and some day something will come up to remind me of it. These damper regulators, for example, which we can adjust from either the engine room or the boiler room, came into my mind when we were having trouble with the smoke inspector. I asked him at the time what he thought of them, and we looked up together an advertisement I remembered having passed over repeatedly. Afterwards he mentioned the apparatus to other concerns around here and I know quite a number that have put it in.

"Oh, you bet, these ads are mighty handy in keeping us up to date. I'm a good deal better engineer than I would be if we didn't have them to refer to right along."

And, personally, I could say the same thing of my own work.

NEW EQUIPMENT

ATLANTIC COAST STATES

Sargent & Munch, Hinsdale, Mass., will construct an electric-light plant at Hinsdale.

Bids will be received by R. S. Craig, until June 10, for the construction of an electric-lighting system in the court house at New Rockford, N. Y.

T. E. McGarr, secretary of the state hospital commission, Albany, N. Y., is receiving bids for refrigerating and ice-making machinery to be installed in the cold-storage plant at the Utica State Hospital, Utica, N. Y.

Rahway, N. J., contemplates the construction of an electric-light plant.

The borough council contemplates the construction of a municipal water-works system at Coudersport, Penn.

The Philadelphia, Baltimore & Washington R.R. Co. contemplates the construction of an electric-light plant at Perryville, Md. J. C. Auten, Wilmington, Del., is division engineer.

SOUTHERN STATES

Bids will be received by Guy Cushman, Eleventh Cavalry, Construction Quartermaster, until 11 a.m., June 15, for the construction of an electric-light system at Fort Oglethorpe, Ga.

Bonds for $35,000 have been voted for the completion of the water-works system and the installation of an electric-light plant at Jesup, Ga.

The Consolidated Ice & Power Co. contemplates improvements to its plant at Valdosta, Ga. H. T. Hartman, Philadelphia, Penn., is interested.

John T. Hammond contemplates the construction of an electric-light plant at New Smyrna, Fla.

R. C. Huston, Exchange Bldg., Memphis, Tenn., is preparing plans for the improvement and extension of the water-works system at Tuscaloosa, Ala.

The substation of the Alabama Power Development Co., on Court St., Talladega, Ala., was destroyed by a cyclone, May 8. Estimated loss, $65,000.

Important improvements are to be made in the plant of the Arctic Ice Co., Chattanooga, Tenn. E. O. Wells is general manager.

The Mt. Sterling Water, Light & Ice Co. will soon start work on the construction of a filtration plant at Mt. Sterling, Ky. L. G. Kirkpatrick is superintendent.

CENTRAL STATES

The installation of an electric-lighting system at Oak Harbor, Ohio, is under consideration.

Alexandria, Ind., contemplates the installation of an electric-light system.

Pigeon, Mich., contemplates the installation of a municipal electric-light plant.

The Rock River Power Co., Byron, Ill., will construct an electric plant to cost $20,000. G. M. Clark is president.

Bids will be received by the city council until July 1, for furnishing equipment for an electric-light system, at Champaign, Ill.

Charles Cottingham, Danville, Ill., has been retained to prepare plans for the construction of a water-works system at Georgetown, Ill. Estimated cost, $10,000.

The Collinsville Electric Light & Power Co. contemplates the construction of an electric-light system at Marine, Ill. A. W. Crawford is vice-president.

WEST OF THE MISSISSIPPI

The power plant of the Waterloo, Cedar Falls & Northern Ry. Co., at Waterloo, Iowa, was destroyed by fire, May 7. Estimated loss, $125,000.

Plans have been prepared by T. Archer & Co., Kansas City, Mo., for extensions to the street-lighting system in Larned, Kan.

ENGINEERS WANTED

Under this head are published, without charge, advertisements of actual vacancies in power plants for operating engineers and assistant engineers. For the guidance of applicants for advertised positions, the advertisements must contain particulars to show the class of service demanded and the wages paid. Employers who desire to advertise under this head are requested to write for a blank form of application.

IN TROPICAL COUNTRY under American jurisdiction, want watch engineer, unmarried man, who can operate 10,000-hp. Parsons turbine plant, a.c. machinery, with centrifugal pumps and other auxiliaries, and who can take down Parsons turbines for repairs; work eight hours per day; salary $125 per month and traveling expenses. Box 675, Power.

HELP WANTED

Advertisements under this head are inserted for 25 cents per line. About five words make a line.

SELLING AGENTS wanted in important centers for Betson plastic firebrick; see page 138 in this issue.

SALESMAN — Thoroughly competent steam specialty salesman; one that can sell high-grade goods. Address "M. M. Co.," Power.

STOKER ERECTORS, first class; none but experienced men need apply; state age, salary wanted, habits and references. Box 668, Power.

THE VULCAN SOOT CLEANER offers an exceptional opportunity for power specialty salesman Address G. L. Simonds & Co., 802 Steinway Bldg., Chicago, Ill.

AGENTS WANTED to sell forced draft blowers and damper regulators; will give exclusive territory to right parties; commission on sales. Turbo-Blower Co., 30 Church St., New York.

AGENTS selling to steam plants can make big money selling for us; no samples; no experience is necessary. Armstrong Mfg. Co., Springfield, Ohio.

MANUFACTURERS' AGENTS to handle thoroughly guaranteed steam specialties; feed water heaters, steam and oil separators, traps and exhaust pipe heads; not necessary to represent entire line if interested in only part of it; our specialties are well advertised and of good reputation. Box 569, Power.

SALESMEN—We need more local sales representatives; we manufacture a specialty for water-works and industrial plants; a valuable agency for energetic men; commission only; now open, Vancouver, Edmonton, Montreal, Pennsylvania, most of the southeastern states and far southwestern states; we prefer steam specialty salesmen now handling other lines or engineers with sales ability. Box 669, Power.

WOULD ONE HUNDRED DOLLARS per week interest you? Would you like to be outside in the fresh invigorating air during the coming months? Free to do your own bidding and earning $100 or more per week, as you choose. Your own boss. We want one first class salesman to represent us in every city in the United States, to sell our high grade specialties to engineers; exclusive territory given. Strong Machinery & Supply Co., 48 Franklin St., New York City.

SITUATIONS WANTED

Advertisements under this head are inserted for 25 cents per line. About five words make a line.

SUPERINTENDENT—Advertiser wants position to take charge of mechanical department and power plant. Apply Box 673, Power.

ARE YOU THINKING of installing new motive power or rearranging your plant? If so, let me help you. Write for particulars. Box 674, Power.

BY A COLLEGE ENGINEER with over six years' experience in running and setting up of steam, gas and gasoline engines of different kinds; good habits; 31 years and single. Box 672, Power.

PRACTICAL ENGINEER, 30 years of age, married, steady and sober, with 10 years' experience in several steam plants up to 2500 hp., with automatic, simple and cross compound Corliss condensing and noncondensing engines, a.c. and d.c. generators and motors; good mechanic with shop experience, desires position as chief in medium or assistant in large plant. Address Box 671, Power.

MISCELLANEOUS

Advertisements under this head are inserted for 25 cents per line. About five words make a line.

PATENTS SECURED—C. L. Parker, Patents, 904 G St., Washington, D. C.

PATENTS—Herbert Jenner, patent attorney and mechanical expert, 608 F St., Washington, D. C. I make a free search and report if patent can be had and the exact cost.

WILL PAY 35 cents per copy for "Power" (monthly), issued January, 1908, May, 1909, and "Power," Jan. 16, 1912. Address H. M. Howell, 1730 Laguna St., San Francisco, Calif.

ENGINEERS AND FIREMEN—Send 10 cents in stamps for a 40-page pamphlet containing a list of questions asked by an examining board of engineers. Stromberg Publishing Co., 2703 Cass Avenue, St. Louis, Mo.

EVERY ENGINEER should be posted regarding the new system of vacuum heating installed without payment of royalty; I have valuable information: write today. M. Y. C., 1417 W. Jackson Blvd., Chicago, Ill.

FOR SALE

Advertisements under this head are inserted for 25 cents per line. About five words make a line.

FRANKLIN WATER TUBE BOILER, 182 hp., used three weeks. Franklin Boiler Works Co., Troy, N. Y.

ONE 120-kw. a.-c., 440-volt Westinghouse generator. One 16x42 Allis Corliss engine, complete with all fittings, including exciter and switchboard. Address "Engineer," Box 2, Station A, Cincinnati, Ohio.

350 KW. WESTINGHOUSE direct current generator, 250 volts, direct connected to 24x36 heavy duty Corliss engine; unit complete with switchboard, balancing set, etc. Arthur S. Partridge, 417 Pine St., St. Louis.

75-KILOWATT gas producer plant; after two years of service, discontinued on account of increased power requirements; complete with all auxiliaries; price very low; quick sale is desired. For particulars address Box 670, Power.

35-KW. GENERAL ELECTRIC generator, 120 to 125 volts, type M. P., Form L, with field rheostat, direct connected to 12x12-in. Fischer engine, 300 r.p.m., all in good condition. Can be seen running in pressroom of Lynn Item, Lynn, Mass.

SWITCHBOARD, capacity 150 kilowatts, d.c., equipped with double busbars, two Weston voltmeters, two ammeters, two I. T. E. circuit breakers, twenty circuit switches. Stephen Greene Co., Arch and Sixteenth Sts., Philadelphia.

CORLISS ENGINES: 5000 hp. Allis vertical cross compound Corliss; 1200 hp. Hamilton horizontal cross compound Corliss; 750 hp. Filer & Stowell cross compound horizontal heavy duty with condenser. Arthur S. Partridge, 417 Pine St., St. Louis.

500 HORSEPOWER right-hand engine; simple Corliss heavy duty frame, size, 24x48, complete, including flywheel, 18 ft. diameter, grooved for 14 turns 1¾-in rope, American drive-steam receiver separator, 7-in. steam pipe including one long radius bend and flanges;

can be seen in operation at our mill in Mt. Carmel, Ill. Address Bernet Craft & Kauffman Mlg. Co., Office 1036 Pierce Building, St. Louis, Mo.

HALF OR ALL of an established machinery business in first-class location as to trade center in a Western state; warehouse and office convenient to center of city of 100,000, on transcontinental railway; own buildings; long lease on ground; reasonable terms; specialize wood-working machinery, transmission, supplies, power equipment and pumping plants; splendid lines of agencies; buyer must have $50,000 to $100,000; none but principals need apply. Address "Bargain," 665 Power.

ONE HEWES & PHILLIPS Corliss engine, 14x36 in., in excellent condition; flywheel, 10 feet diameter, 18 in. face, five to six years old. One 75-kw. a.c. Westinghouse generator, 3 phase, 220 volts, 60 cycles, 900 r.p.m. with switchboard and exciter complete. One 50-kw. a.c. Westinghouse generator, 3 phase, 220 volts, 60 cycles, 1200 r.p.m. with switchboard and exciter complete. One 45-kw. d.c. General Electric generator, 115 volts, 850 r.p.m., with switchboard complete. The Salt's Textile Mfg. Co., Bridgeport, Conn.

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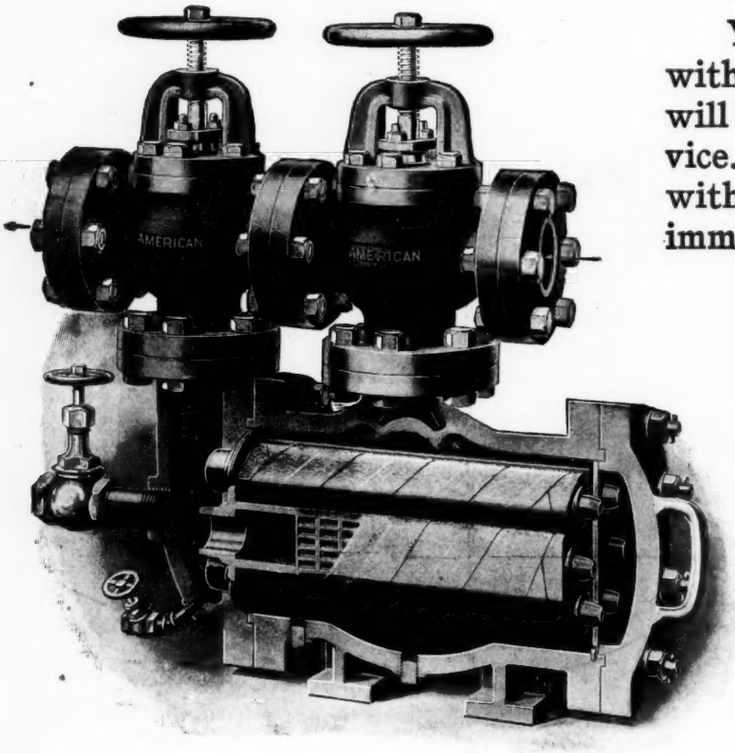

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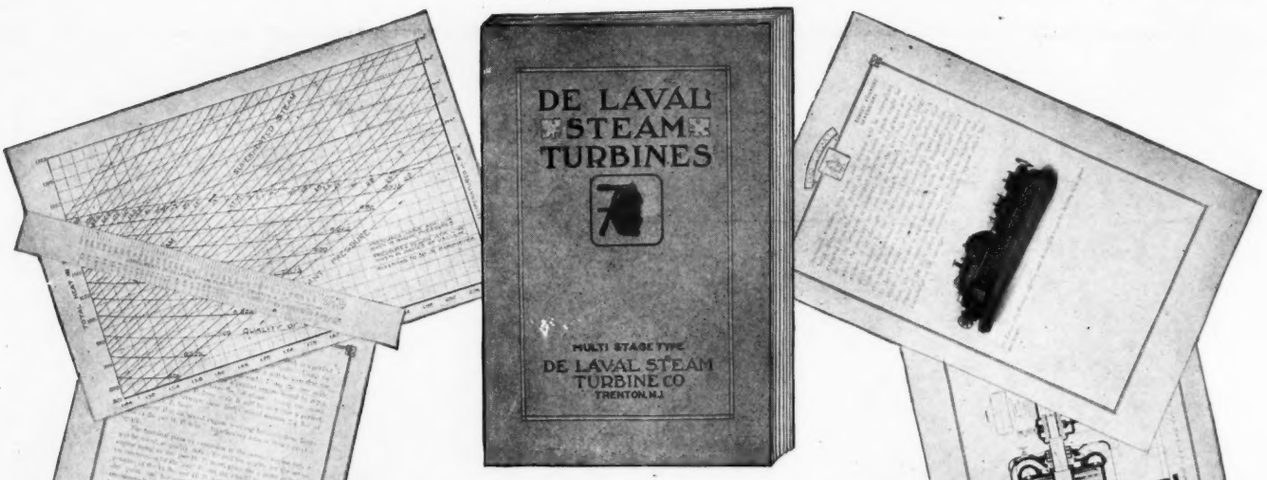

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